COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

l. 8, No. 9 MARCH, 1937 MAR 25 1937 25c a copy

Photo by Doris Day

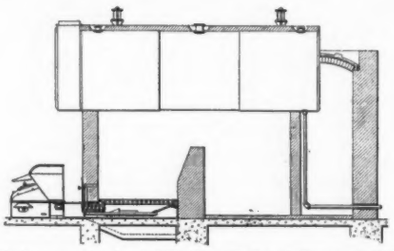
Windsor Steam Plant

Heater Closures

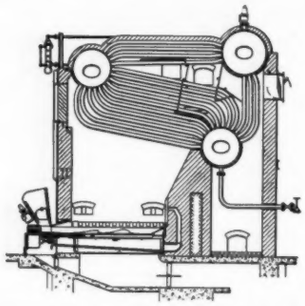
Steam or Electric Drive?

Regulation of Feedwater Level in Boilers

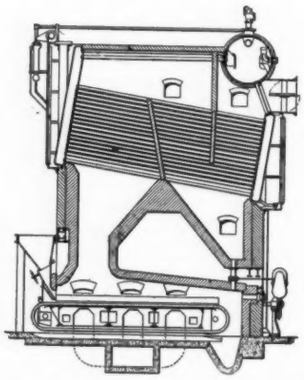
MODERN STEAM GENERATING UNITS for small and medium-sized plants



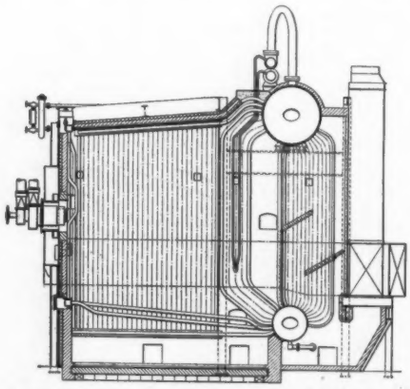
Unit comprised of CE Fire Tube Boiler and CE-Skelly Stoker Unit. Boiler h.s.—1525 sq ft Similar units are available in sizes down to 300 sq ft



Unit comprised of CE Bent Tube Boiler (Type VM) and CE Underfeed Stoker. Boiler h.s.—2250 sq ft



Unit comprised of CE-Heine Box Header Boiler and CE-Coxe Traveling Grate Stoker. Boiler h.s.—5000 sq ft



CE Steam Generator (Type VU). Maximum continuous output—75,000 lb per hr. Units of this type are available for capacities from 20,000 to several hundred thousand lb of steam per hr

The accompanying drawings of units installed or on order are representative of CE installations in small and medium-sized plants. Each of the various types of boilers and firing equipment is available in a range of sizes adequate for the conditions to which it is applicable. These products, in conjunction with others, comprising the complete CE line, enable Combustion Engineering to supply the most suitable and satisfactory types of equipment for any combination of load and fuel conditions.

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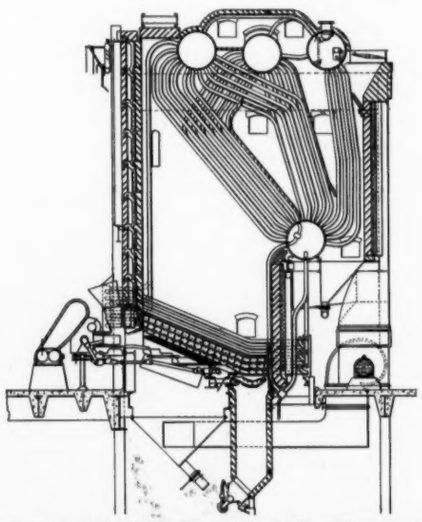

Unit comprised of CE Bent Tube Boiler (Type VA) and CE Multiple Retort Stoker. Boiler h.s.—5000 sq ft

COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

VOLUME EIGHT NUMBER NINE

CONTENTS

FOR MARCH 1937

FEATURE ARTICLES

Windsor Steam Plant Modernization	*by Philip Sporn*	20
Auxiliaries—Steam or Electric Drive?	*by Theodore Baumeister, Jr.*	25
Heater Closures	*by George A. Fricke*	31
Regulation of Feedwater Level in Boilers	*by V. V. Veenschoten*	34
How Engineers May Profit from Patented Inventions	*by Leo T. Parker*	37

EDITORIALS

Engineering Unemployment during the Depression 19
An Economist Compares Steam and Water Power 19

DEPARTMENTS

Steam Engineering Abroad 39
Advertisers in This Issue 42

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Power

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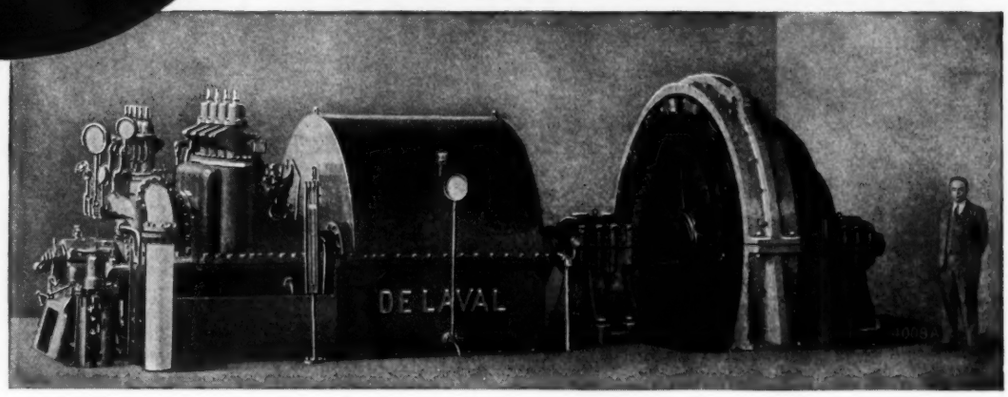

De Laval 4500 r.p.m. extraction turbine geared to 375 r. p. m., 2500 kw., D. C. generator for chemical manufacturing plant.

IMPROVEMENTS IN BOILER ROOM DESIGN AND PRACTICE AND IN

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HAVE CUT BOTH THE FIRST COST OF INDUSTRIAL POWER PLANTS AND COST OF FUEL PER UNIT OUTPUT APPROXIMATELY IN HALF SINCE 1919

Our engineers will be glad to cooperate in the study of any power problem by supplying engineering data and estimates.

Improvements in steam plant performance have resulted from the ability to burn cheaper fuels at higher rates and to absorb through reduced heating surface a greater percentage of the heat developed; also from increased steam pressure, from superior feed heating cycles and from refinements in the design of turbines, which operate at higher speeds and more efficiently.

In redesigning De Laval steam turbines advantage has been taken of (1) high strength alloys and (2) better metallurgical control, of (3) scientific experiments upon, and (4) better materials of construction for, rotors and blading, of (5) streamlined steam flow passages, of (6) means for separating moisture from the steam, and of (7) more accurate and more stable governing devices.

In some cases, greatly increased power can be generated from the same amount of fuel, without displacing existing turbines, by super-imposing high pressure boiler and turbine units. In other cases, power is generated at almost no cost for fuel by installing back pressure or bleeder turbines.

3188

De Laval *Steam Turbine Co.*, Trenton, N.J.

Manufacturers of Steam Turbines, Centrifugal Pumps, Propeller Pumps, Rotary Displacement Pumps, Centrifugal Blowers and Compressors, Worm Gears, Helical Speed Reducing Gears, Hydraulic Turbines and Flexible Couplings. Sole Licensee of the Bauer-Wach Exhaust Turbine System.

EDITORIAL

Engineering Unemployment during the Depression

The plight of engineers during the depression was everywhere apparent but the extent to which those in the profession suffered from unemployment could be judged, at the time, only by personal observations, by applications filed with employment agencies, and by the marked falling off in engineering society membership. Now, however, concrete figures are available, for the results of a survey conducted by the U. S. Bureau of Labor Statistics, covering the five-year period of 1930 to 1934, inclusive, are contained in the Bureau's *Monthly Labor Review* for January 1937. While the figures, obviously, do not represent a one hundred per cent coverage of the profession, they have been compiled from questionnaires returned by over fifty thousand engineers and therefore may be taken as representative of conditions that obtained during the worst years of the depression.

It appears that, although at no time were more than eleven per cent of the engineers unemployed in some capacity, more than one-third of the total reporting were out of employment at some time during the depression. The periods of such unemployment varied from an average of about a year for the younger engineers to nearly two years for the older men. This may be partly accounted for by the fact that many of the younger men sought and obtained employment in fields other than engineering while the older men were reluctant to take up new lines. From an analysis of the figures it would appear that employment among the experienced engineers followed more closely the activity in the capital goods industries. The older men felt the depression earlier and at the end of 1934 were still lagging. College graduates were slightly better off than those without degrees.

There was no marked difference in unemployment as between different groups of engineers, except that by 1934 the conditions of chemical and civil engineers had improved, the latter probably because of the public works that had got under way by that time.

It is significant that while four per cent were on work relief, less than one per cent received direct relief from the Government. The latter percentage may be somewhat misleading inasmuch as many engineers received direct relief from agencies within the engineering profession. Compared with the general public, however, these figures would indicate that engineers as a class had accumulated some resources that enabled them to subsist during periods of unemployment.

Such statistics are always interesting, but the unavoidable delay in their collection, compilation and analysis render them of little value in establishing corrective measures while the emergency exists. They merely serve now to confirm opinions formed by close observers. Perhaps they may be useful as the basis of studies aimed at improving the future security of the engineer, but the fortunes of the profession are affected by so many factors beyond its control that this remains doubtful. Had the survey revealed the change in the economic status of engineers, as brought about by the depression, it might have served a more useful purpose. Perhaps this information will be forthcoming in a later study.

An Economist Compares Steam and Water Power

Raymond Moley, one of the original so-called "brain trust" advisors to President Roosevelt, in a special article in *Today*, of which he is editor, compares the relative costs of water power and steam power, and takes issue with the popular conception that water power is analogous to cheap power—a conception which appears to be shared by the Administration at Washington.

Reviewing the steady increase in the efficiency of steam power plants and the possibility of further improvement through new developments, Professor Moley points out that, not only has water power long since reached its maximum efficiency, but that all the choice water power sites have been developed and most of those remaining are not within the range of economic transmission distance to load centers. Water power capacity costs, in general, about three times that of steam capacity and, despite the cheapness of capital when supplied by the Government, the high initial cost offsets the absence of fuel cost, not to mention the increased transmission costs.

Another point raised by Professor Moley is the possible effect on the coal mining industry through the program of extensive water-power development which the Federal Government has undertaken. Quoting statistics from a World Power Conference paper dealing with our coal reserves which, allowing for growth of consumption, appear adequate for a period of 4000 years, he concludes that conservation is not a pressing factor and that these ample reserves, plus the constantly reducing cost of steam production, "put off the possible need of developing water power to a time wholly beyond our present concerns."

"Private enterprise," he says, "believes its future lies in moderate-sized generating plants and shorter transmission lines, that one of the cheapest ways to transmit power is to haul it by rail in a coal car, and that a government whose objective is only the insurance of cheap and abundant power for its people might do worse than to consider such things as these."

While all this is familiar to well-informed engineers, it is gratifying to find one in Professor Moley's position having such a thorough grasp of the economics of the power problem. He might also have added that, with few exceptions, dams and reservoirs constructed for flood control have a function quite opposite to that of water storage for power generation.

Would that he were still high in the councils at Washington.

Windsor Steam Plant Modernization

By PHILIP SPORN

Vice President and Chief Engineer
American Gas and Electric Co.

A further advance in the development and application of superposition

The topping installation that is now going in at the Windsor Station consists of two 750,000-lb per hr units supplying steam at 1250 lb and 925 F total temperature to a 60,000-kw, 3600-rpm turbine-generator exhausting to three existing low-pressure turbine-generators of 90,000 kw combined capacity. One of the steam-generating units will have a slagging type of furnace and the other a dry-bottom type. Each employs different and novel means of superheat control.

SINCE the formulation and statement of the basic ideas behind the Logan superposition installation,[1] a great many other topping installations have been undertaken and are now under way. Among these are Waterside, Springdale, West End, Fiske Street, Schuylkill, Dayton and a number of others. Basically they all follow the Logan pattern, i.e., the pressure is of the order of 1250 lb, the steam temperature is around 900F and they use a unit of from 30,000 to 50,000 kw for topping purposes. In boilers, Logan's lead in the direction of one boiler has not been followed so directly. Rather, a more conservative practice, usually with two boilers, has been generally employed for a topping capacity of from 30,000 to 50,000 kw. So far as is known, all the topping machines of 60 cycle rating employ a speed of 3600 rpm, and those of a rating of 30,000 kw or over, employ hydrogen as a cooling medium. But basically, the same ideas developed for Logan have been employed in all the subsequent installations.

The Windsor superposition, however, carries forward a number of the ideas previously employed. As was true in the case of Logan, if one had freedom of choice, the choice would naturally be in the direction of saying not too much, if anything at all, about what is being planned. But apparently the idea of saying nothing in connection with the developments that have as much technical and economic interest as some of those under discussion, is possible in modern times only in the case of laboratory developments. At any rate, because some of the ideas being developed at Windsor have already been the subject of considerable informal discussion, it appears desirable to make some authentic statement. Further, it is hoped that such a statement and the knowledge of what is being attempted at Windsor, will stimulate further constructive thinking.

It may be of interest to review the salient points of the new ideas and equipment developed for Logan:

In the turbine a size of unit was undertaken that was at that time over twice the size of the largest unit previously attempted at 3600 rpm; the throttle temperature was the highest attempted on a regular commercial unit of U. S. manufacture. The turbine, too, was the first to employ the double-casing construction on the steam end for handling the high-temperature stress problem. The generator embodied not only new principles and new materials in the construction of the rotor through the use of aluminum windings, but was the first commercial alternator to be hydrogen-cooled. The stator employed a new method of cooling the iron and the embedded portion of the armature coils, and for the first time hydrogen was used for the cooling of the rest of the winding. A totally new method of sealing against hydrogen leakage was employed in the Logan machine.

The Logan boiler was the largest boiler attempted in that temperature and pressure range and was definitely laid out with the idea that service availability would be high enough to make such a size economically feasible. As against the more prevailing practice of handling the ash in the wet condition, the Logan boiler, because of the broad load range that had to be handled, was definitely designed for a dry bottom. Further, the rate of heat liberation and the radiant absorbing surfaces were definitely laid out with the idea of making the upper reaches of the boiler substantially free from slag deposits under all operating ranges up to full load.

The cycle at Logan was notable not only for its direct and simple flow arrangement but for the unusually high ratio between high-pressure and topped capacity, the ratio developed being 1.16 against the more common one of roughly 2 to 3. This was accomplished by dropping the cross-over pressure and by the introduction of turbine-drive for auxiliaries that had heretofore become more or less standardized on an electrical-drive basis.

On the electrical end a more or less complete divorce of the new unit from the electrical layout of the old plant was accomplished. An integration in a single transformer bank of all the voltages serving this station, giving all the necessary synchronizing power between the various units of the station without in any way upsetting the short-circuit limits for which old parts of the station, had been developed.

In reviewing the foregoing high lights of Logan, it is well to keep in mind that there were a number of problems that had not been completely solved. Among these were the floating throttle valves on the turbine.

[1] *Electrical World*, April 11, 1936.

These admittedly introduced complicated problems in one behavior of turbine and turbine piping. Again, the work carried out on the development of the Logan turbine definitely showed that the various design principles and ideas employed in this unit could be satisfactorily projected into sizes well beyond the Logan size. Hence, there was a natural conclusion to the effect that where a proper opportunity presented itself, the development of a larger topping unit would make possible very attractive economies. On the boiler end, several problems which were satisfactorily solved at Logan had solutions that did not have general application. Thus, the dry bottom, it was felt, might present many difficulties in the handling of coals having much lower fusion temperatures than those encountered at Logan. Again, it was felt that the solution of the problem of superheat control at Logan was only approximate and that further development of higher temperatures and pressures would call for a closer and more definite working out of superheat regulation.

The Problem at Windsor

The Windsor Plant was originally erected in 1916 and extended to its present size in 1922. Since that time it has been operating at its present size as a six-unit plant, with six 30,000 kw turbines, four of 250-lb throttle pressure, and two of 325-lb throttle pressure, and at a steam temperature of approximately 625 F. The Windsor Plant has thus at the present time an effective life of close to twenty years. It is an interesting digression that within the last year the Windsor Plant has established not only the lowest Btu performance in its history but also, in spite of increased material and labor costs, the lowest cost per kilowatt-hour in its history. Nevertheless, it was definitely recognized that the limit to what could be done in the way of economies, thermal or dollar-wise, had definitely been reached and that with further age, performance, of necessity, would go down. Thus the conclusion was reached that under the normal, but at present particularly intense, pressure to reduce costs something more radical than mere tightening, so to speak, of the old plant would have to be done and superposition began to be studied.

The first studies undertaken in 1933 were handicapped in an economic sense by the fact that at that time the maximum size turbine for a superposition unit that appeared feasible was of 38,000 kw rating. This called for three topping units to handle the old plant. Besides, it made a rather awkward arrangement for topping since the Windsor Plant is a jointly owned property, one-half being owned by The Ohio Power Company and the other by the West Penn Power Company. A maximum topping unit size of 38,000 kw meant, therefore, joint ownership of one of the topping units. This, together with the abnormally low loads then being carried on all utility systems, made it difficult to show an adequate economic justification for proceeding with the work when additional capacity seemed of no immediate value on the system.

Meanwhile, turbine and boiler designs were making rapid progress. In stride with this, the Logan development was carried through. The work done there, coupled with some of the other work carried out on other installations, definitely showed the way to a more economical plan of development at Windsor. This, together with the growth in load and the need of additional capacity, made superposition much more attractive and cleared the way of all difficulties that had previously hindered it. The high lights of that development are here described.

Although the West Penn Power Company has under construction a turbine exactly similar to the one described below, the other parts and arrangements described, such as boilers, heat cycle, etc., apply to The Ohio Power Company's half of the plant only, the West Penn Power Company's half of the plant being scheduled for construction at a later date.

The Turbine-Generator

The turbine will be a twelve-stage unit rated 60,000 kw at 90 per cent P. F. and operating at a speed of 3600 rpm. It will have a throttle pressure of 1250 lb, a

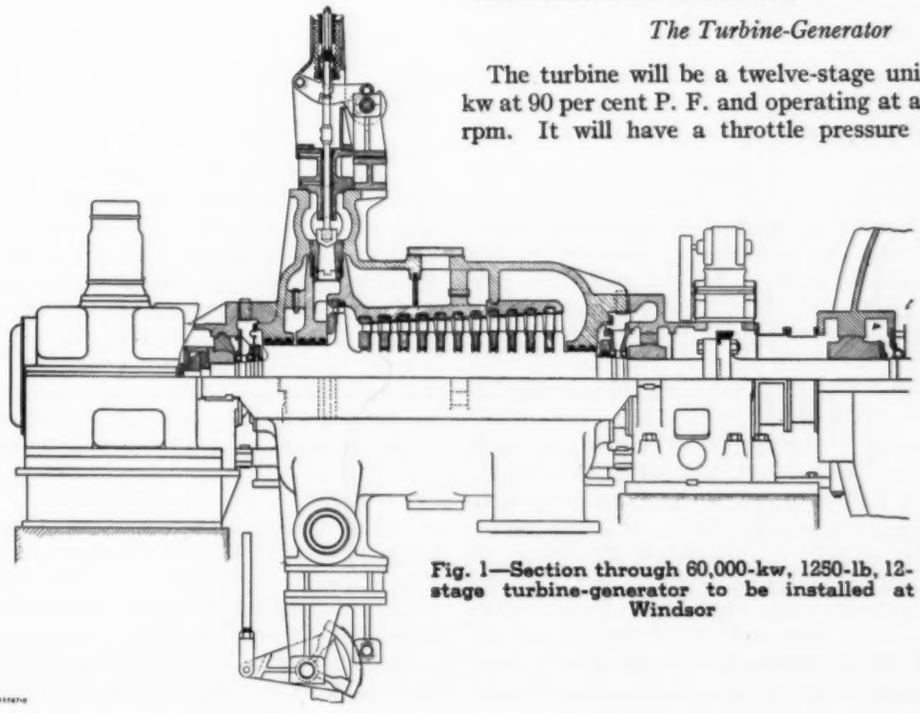

Fig. 1—Section through 60,000-kw, 1250-lb, 12-stage turbine-generator to be installed at Windsor

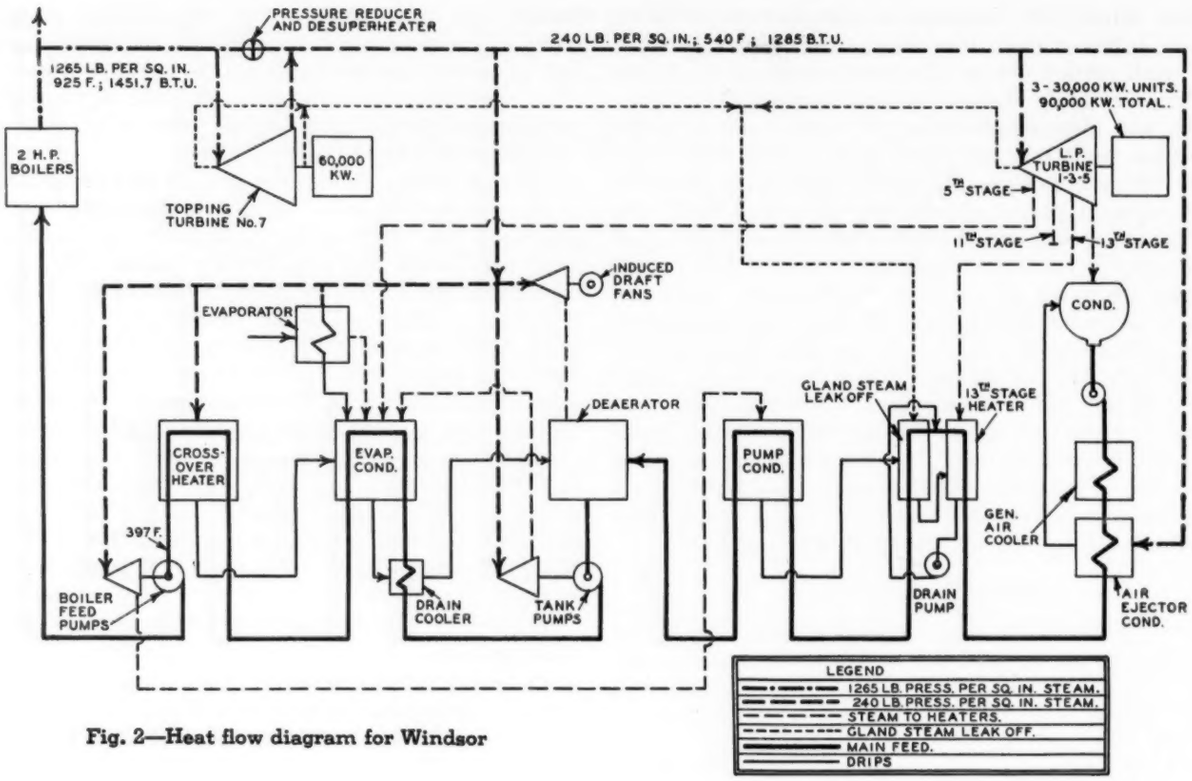

Fig. 2—Heat flow diagram for Windsor

throttle temperature of 925 F and exhaust against a back-pressure of 235 lb gage.

In size the unit will thus be larger than any that has heretofore been attempted in a superposition. The need for such a size, as has already been mentioned, is the divided ownership of the plant. Thus, if ownership of one of the high-pressure units was not to be mixed it was a question of topping one-half of the plant with either one or two turbines. The additional economies that one turbine offered, plus the satisfactory development of the design work on the Logan unit, prompted the decision to go up to the 60,000 kw size.

As will be seen in Fig. 1, showing a cross-section, the turbine uses the by now well known double-casing construction. The throttle valve, however, is definitely anchored to the turbine foundation and this, it is felt, is an advance in turbine design. Further, as contrasted with the Logan unit, in which the distributing valves were all located below the turbine floor line, the Windsor turbine has valve-in-head construction, thus considerably simplifying the piping problem both from a construction and an operating standpoint. The generator itself is, in practically all respects, substantially similar to that employed in the previous design for Logan. It is designed for 11,200 volts, but with a lower short-circuit ratio of 0.85. The system at Windsor is such, however, that this short-circuit ratio value gives ample margin of safety from a stability standpoint. As contrasted with Logan, where a new system of stator cooling by use of "water pads" was introduced, the system at Windsor will employ the conventional stator construction utilizing four banks of fin-type coolers located within the stator for that purpose. This change was due not to any difficulties encountered in the development of the Logan design, but simply as a measure of conservatism while awaiting actual operating results with the unusually novel stator construction at Logan. Again as in the Logan design, the exciter will be direct-connected and geared, but the excitation system will be somewhat simpler, the high-pressure unit supplying merely its own excitation and the low-pressure units continuing to supply their excitation from their own direct-connected exciters as at the present time.

The Heat Cycle

The cycle chosen is in all but a number of minor respects similar to that adopted for Logan. As will be seen from an examination of Fig. 2, which shows a heat flow diagram, two high-pressure boilers supply the necessary steam for a 60,000-kw high-pressure unit which, in turn, exhausts and supplies three low-pressure turbines having a combined rating of 90,000 kw. The pump condensers have been advanced somewhat in the cycle, primarily for mechanical reasons, but in all other respects, and particularly as regards simplicity, use of turbines for boiler feed and induced-draft fan drive and rather limited bleeding, the cycle follows the Logan setup. The expected performance on full load under normal conditions will be better than at Logan, due to better Rankine efficiency of the Windsor low-pressure units and their more efficient low-pressure cycle.

The Boilers

Two 750,000-lb per hr boilers will supply the steam at a pressure of 1350 lb and a temperature of 925 F for a total capacity of 150,000 kw. The decision to use two boilers was influenced by the requirements of 1,500,000 lb steam per hour and the limited experience to date with boilers even of 1,000,000 lb rating, at 1350 lb pressure, such as the Logan design. A single boiler would

definitely involve experimentation with too large a number of unknown factors. On the other hand, it was felt that nothing beyond two boilers would be justified in view of the satisfactory design development of the Logan boiler and other boilers of which it was a forerunner.

It will be noted from Fig. 3, which shows a cross-section of each of the two boilers, that two considerably different designs were chosen, although they will be similar externally. Thus, the boiler on the right is a three-drum, so-called VS-3 type of unit employing tangential firing, very conservative heat release rates, a steeper than normal angle for the bent-tube vertical sections, a convection type superheater and standard arrangement of economizer and air heater. The forced-draft and induced-draft fans and air heaters are located on a common topmost level. The bottom is designed for continuous slag removal. Three Raymond bowl type mills are employed for the pulverizing. It will be noticed that a system of auxiliary burners to be supplied either from the main mills or from an additional mill, depending upon details yet to be worked out, are designed for front wall firing at a level just above the mud drum. This, together with damper bypass, will give a degree of superheat control over a wide range impossible with damper bypassing alone.

The second boiler, of a different manufacture, designed within the same space limits, uses a three-drum, center-split furnace arrangement (see Section A-A) with a very limited straight tube section. The two halves of the split furnace are independently fired, one-half being equipped with a radiant-type superheater in the wall section. The net effect of this is not only to give maximum ease of control of superheat over the entire range desired and within the limits desired, but what is equally important, gives a design that naturally has a larger than normal amount of radiant heat absorbing surface. This at one time not only facilitates the problem of keeping the upper reaches of the boiler from slagging, but makes possible the development of the design for dry-bottom operation even under the rigid operating specifications of fusion temperatures of coal running from 1900 to 2700 F. During starting, the half of the furnace having the radiant superheater is not fired, thus protecting it against overheating. The pulverizers for this boiler consist of three Hardinge type units.

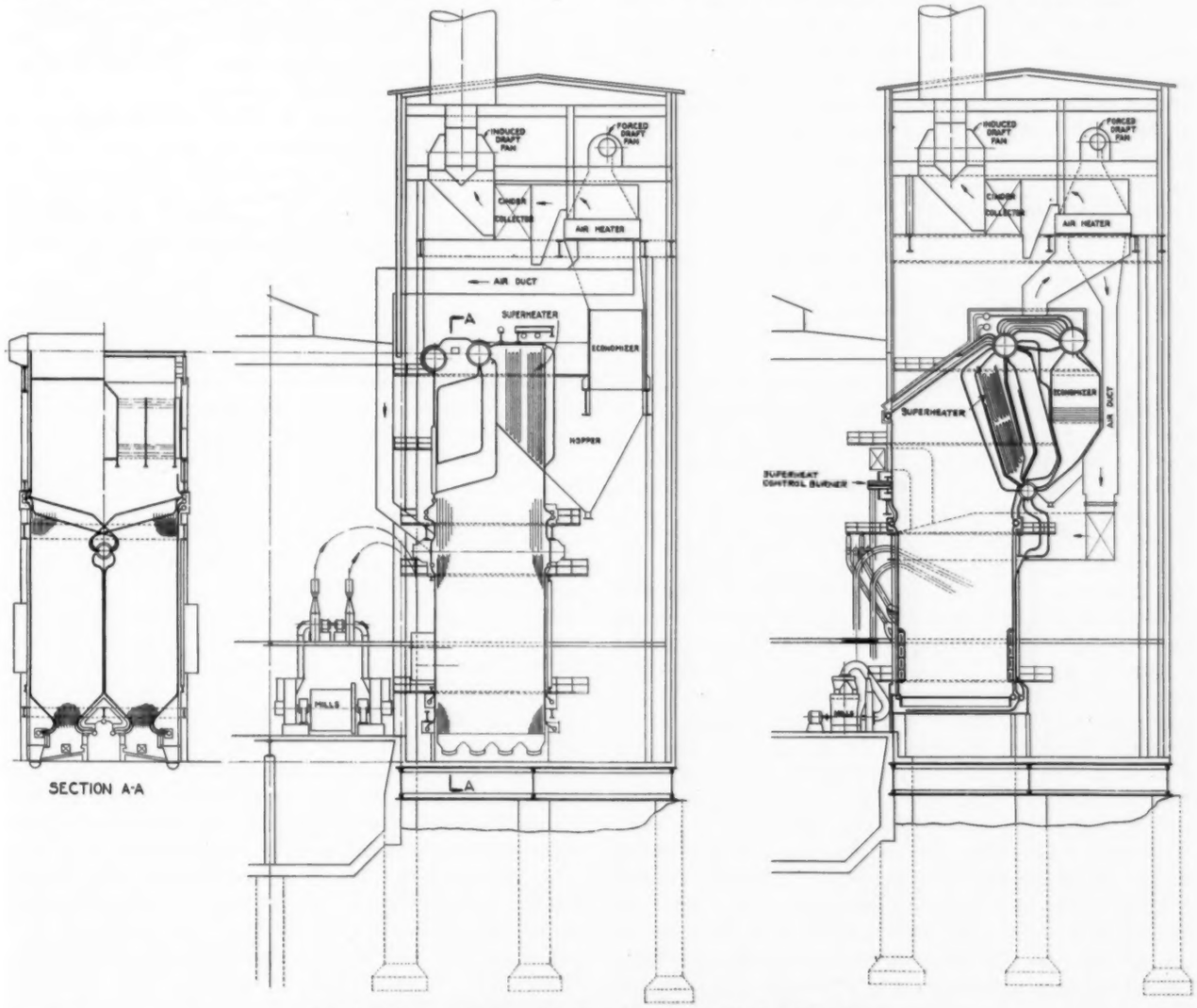

Fig. 3—The two types of boilers to be installed at Windsor

Although similar externally and within substantially the same physical dimensions, two types of boilers considerably different from each other will be installed. That on the right is a tangentially fired VS-3 wet bottom boiler arranged for continuous slag removal. That on the left uses a split furnace arrangement with combination radiant and convection type superheaters. The bottom will be dry for the two halves. In combination, the two boilers will give an unusually wide range of superheat control and a very broad load range from the standpoint of ash and slag handling.

In the development of the Windsor design, the problem of wet and dry operation was given a great deal of thought and the final decision was made to adopt a dry bottom for one boiler and a wet bottom for the second boiler. It has always been recognized that the dry bottom gave the utmost flexibility in range of operation. The difficulty has heretofore been that in the attempt to do a sufficiently large percentage of the work in the water-wall section of the boiler and to keep temperatures at the bottom at a point where no difficulty would be experienced with fused slag, a low superheat temperature may result. The independently fired radiant superheater, however, in this case offered the ideal solution. Further, in so far as the wet bottom has any natural inherent advantage, that advantage always comes into play at higher ratings when fusion of ash is never a problem. Thus, what is not possible normally in a single boiler, that is, the obtaining of the advantages of the dry bottom at low load and such advantages as exist in a wet bottom at high load, are entirely possible for a plant as a whole with one boiler operating dry and the other wet.

A further point of interest in connection with the design is the fact that full use is being made of existing foundation caissons and that out of an existing installation of twelve boilers, only three will have to be removed to make room for an additional amount of steam equivalent from a power standpoint to 150 per cent of the present capacity of twelve boilers. Thus, 75 per cent of the present low-pressure boiler capacity will be left available to back up the high-pressure boilers for low-pressure turbine operation. Some of this gain in space has been made at the expense of additional height but it is nevertheless a very striking example of efficient utilization of floor space.

Electrical Arrangement

Fig. 4 shows the existing and proposed electrical arrangement for tying in the capacity of the new high-pressure turbine-generator (unit 7). It will be seen that the present three generators (half of the total of six) switch at 11,000 volts to a ring bus (not shown in the diagram), with reactors interposed between individual units. From this bus system a group of four 30,000-kva transformer banks feed a 66,000-volt bus and another group of two 50,000-kva transformer banks feed a 132,000-volt bus. The entire low tension bus, one-half of which only is shown in Fig. 4, handles a total of six units and has done so over a period of twenty years with entire satisfaction, both from the standpoint of protection and from the standpoint of mechanical stress. However, the existing bus arrangement, although entirely satisfactory at the present time, was laid out for what was at one time considered an ultimate development and did not, therefore, make any provision for the difficulties inherent in any augmenting of the short-circuit capacity on any particular bus section. The electrical development was, therefore, studied with the idea of staying away from the existing low voltage bus and yet maintaining all the necessary strength of a synchronous tie between the new unit and the old units. The scheme contemplated, it will be noted, proposes to do this by connecting the new unit directly to the low voltage winding of a three-winding bank normally connected to the 132-kv system, but

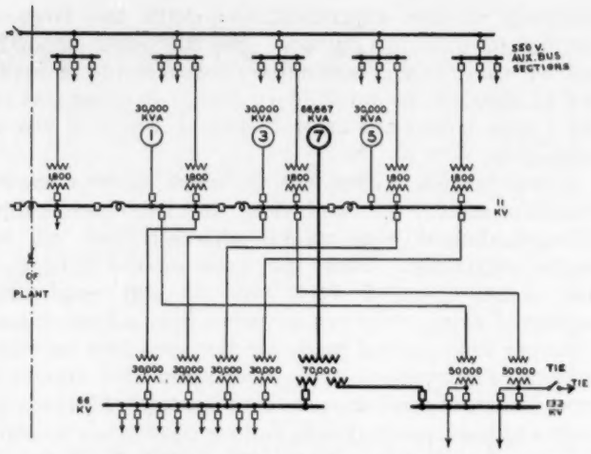

Fig. 4—Simplified one-line diagram of The Ohio Power Company's half of Windsor Plant

Note that the present low-pressure generators are now connected through a reactor sectionalized low-voltage bus. The 66,667-kva superposition unit is, however, not going to be switched to that bus but will be connected normally to the 132-kv bus with an arrangement for a tie in emergency conditionsb etween the high-pressure unit and the 66-kv bus, and between the 66-kv bus and the 132-kv bus.

capable in emergencies, or even under normal conditions, of connecting to the 66-kv system. At the same time a direct low impedance transformer tie is provided between the 66- and the 132-kv systems.

The existing low voltage auxiliary system appears at the present time to be adequate to take care of all the additional new auxiliaries that the new boiler plant will require and this is primarily due to the fact that the two groups of the heaviest auxiliaries consisting of boiler-feed pumps and induced-draft fans are being driven by turbines.

Summary

An analysis of the foregoing short description, particularly if supplemented by a study previously given of the Logan installation, clearly shows that at Windsor:

1. There have been carried forward in a great many respects many of the ideas first developed at Logan. However, a considerable number of new steps have been taken; among these are—

2. The size of the Windsor high-pressure turbine and alternator is considerably larger than any of its predecessors by about 20 per cent.

3. Two different methods for close regulation of superheat will be utilized, each being suited to the particular boiler design or inherently deriving from the particular boiler design.

4. An unusually wide and flexible range of successful station operation has been provided for, not by stretching beyond the natural range either the wet or the dry bottom designs, but combining the two in a manner that sacrifices in neither boiler any of the inherent advantages of the particular type of bottom chosen.

5. Although not touched upon in the body of the article, it is worthy of note that at Windsor there will be attempted the first full supervisory operation of a steam turbine. This is a phase of operation in which steam practice has lagged behind straight electrical and hydraulic practice, but the need for it is as definite in steam work as in the other fields. There is, however, no opportunity further to touch upon this within the confined limits of this article.

Auxiliaries— Steam or Electric Drive?

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The present extensive employment of high-pressure topping installations has revived consideration of steam-turbine drive for certain auxiliaries requiring large amounts of power. In fact, steam drive is being employed in a number of these recent installations. The author analyzes the relative advantages of steam and electric drive for such purpose and presents several cyclic diagrams to illustrate the points raised.

FOR some years it has been prevalent practice to use electrically-driven auxiliaries in large power plants. With the current fashion of topping turbines and boilers for existing low-pressure stations it is pertinent to consider whether the preference for these electric drives is on a sound economic basis. Many designs which are now being effected reveal a trend away from the electric drive and toward the steam drive. This does not include all the auxiliaries in a plant but rather only the major equipment, such as the boiler feed pumps and the induced-draft fans. Prior to 1920 steam drive for auxiliaries was generally favored. However, most of the plants built during the period of 1920 to 1930 used electrical drives almost exclusively. Spare units for such critical equipment as the boiler feed pumps were equipped with turbine drive because of the added reliability. Some plants, which were pulverized-coal fired and interconnected with the power system through a single high-tension line, even went so far as to make this equipment electrically operated, but protected against outage through the medium of a house unit. The house unit was tied in through suitable reactors so that interruption of the main unit operation would not interfere with the major auxiliary drives. These plants, with electrically-driven auxiliaries, all showed good performance records both as to economy and as to reliability. Today the designer has these past records on which to base his choice. In the light of present topping practice, however, it might be well to review the question of auxiliary drive and to re-evaluate the worth of the alternate methods.

Many of the topping installations that are now being built call for very large auxiliaries. One new high-pressure boiler replaces many old low-pressure boilers, and one high-pressure turbine serves several low-pressure turbines with its exhaust steam. Some of these superposed extensions have been justified on the basis of capacity rather than efficiency. The main units and the boilers must handle large quantities of steam and gas. This means that the auxiliaries are of correspondingly large size. With this use of large pumps and fans, under conditions of high head, the drive must be by large power consuming units. A feed pump demanding 2000 bhp or a fan demanding over 1000 bhp is not unusual. Such large size is definitely in favor of steam drive because of the high internal efficiency with increasing turbine size.

Limited Bleeding a Factor in Many Old Low-Pressure Plants

Again, in those plants where superposition is contemplated, there is often a very definite limitation to the bleeding that can be carried out. The existing low-pressure units are often devoid of extraction connections or are so designed as to be capable of only a very limited amount of extraction. When those turbines were built the regenerative cycle had not been brought to its later high degree of perfection. Many existing plants in the 200 to 300 lb group are equipped with a wide assortment of small, inefficient steam auxiliaries. Their exhaust is used for feed heating but their replacement with motors can often be justified, leaving the steam drive for the larger power consuming auxiliaries where variable speed operation can be profitably used. The lack of adequate bleeding facilities has made it necessary for the designers to develop cyclic layouts which meet this limitation. In many of these cycles, the introduction of the steam-driven auxiliary has served to solve the problem. A review of some typical cyclic diagrams reflects several alternate solutions, proposed or in use. Figs. 1 to 5 illustrate a few of these developments.

Fig. 1 is an arrangement which contemplates the use of electrically-driven auxiliaries throughout the plant. The lack of bleed from the main units is offset by the installation of a house turbine. This house unit takes its steam supply from the crossover, usually at 200 to 300 lb pressure, with present 1200-lb topping pressures. It exhausts at a constant back pressure to a deaerating heater in the neighborhood of atmospheric pressure. To gain an additional heating stage between this and the crossover pressure, a bleed point is incorporated on the house unit at 75 to 100 lb. No effort is made to control this bleed pressure and it is allowed to float on the line

†Professor Baumeister, in his capacity as a consulting engineer, has been associated with the design of several of the topping installations referred to in the diagrams shown.—EDITOR

following the characteristics of the house turbine. For heating the feed below the atmospheric boiling temperature, the existing steam-driven auxiliaries or low-pressure bleed points are used. This results in a cycle of good thermal performance, but it offers a definite drawback because of the cost of the electrical connection of the house generation to the main station circuits. This cost burden can be offset by following the procedure terminals of the single topping generator and thus avoids the electrical complications of a separate house unit. This supplementary turbine operates similarly to the house turbine of Fig. 1, and is governed for a constant exhaust pressure of atmosphere plus a small margin.

One bleed point intermediate to the crossover and exhaust pressures is included. The flow through this supplementary turbine is regulated in accordance with

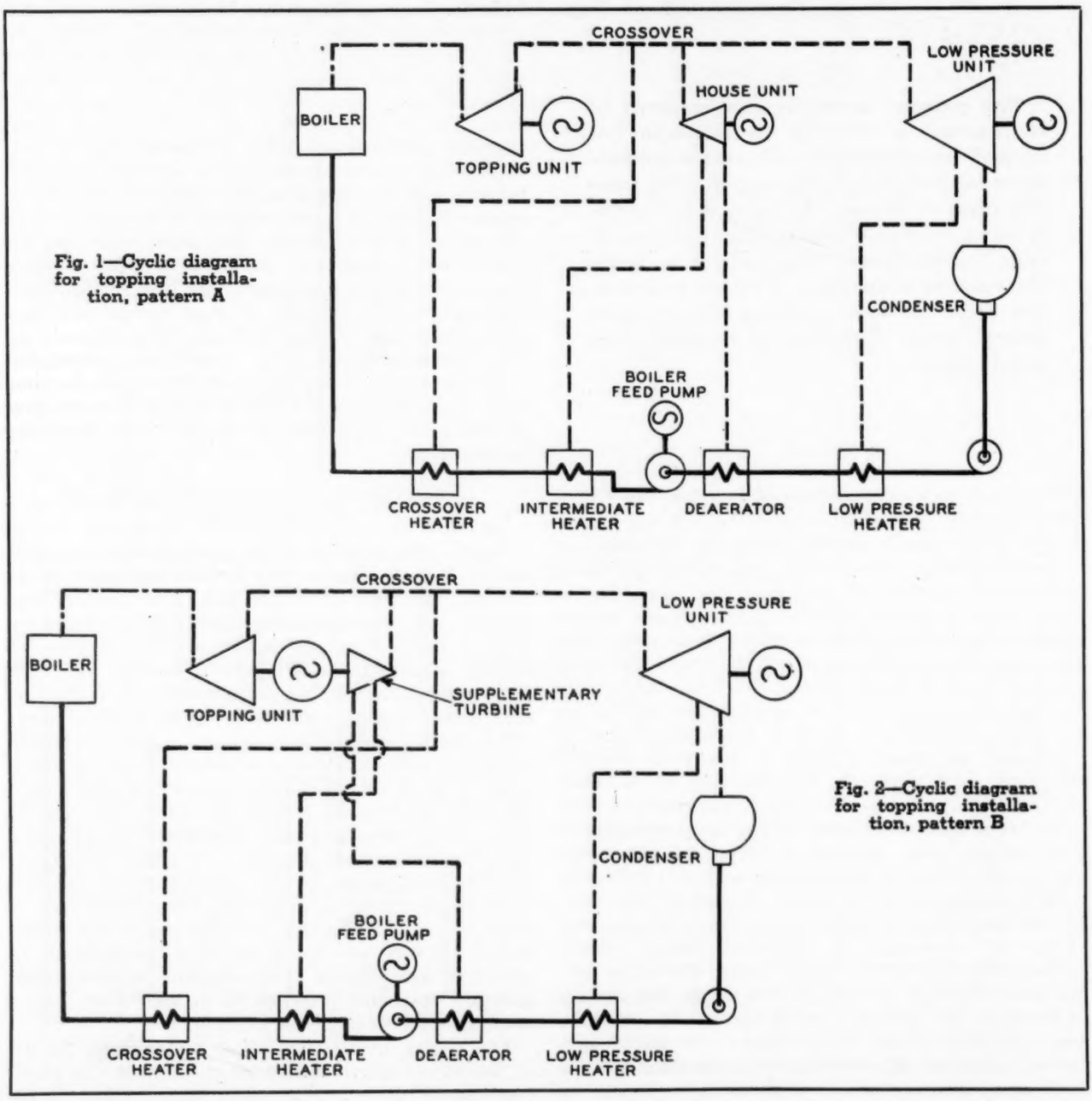

Fig. 1—Cyclic diagram for topping installation, pattern A

Fig. 2—Cyclic diagram for topping installation, pattern B

adopted at Waterside Station and diagrammatically illustrated in Fig. 2.

Here, instead of using a separate house turbine-generator set with its attendant disadvantages, there is added to the high-pressure main unit a turbine which will operate with the requisite steam conditions. This turbine is connected to the main high-pressure turbine-generator shaft so that its output is taken through the the steam demand of the feed system through the action of the back-pressure governor. This supplementary turbine is of high internal efficiency. It consists of only a few stages, and is relatively simple. The flow of steam through this unit is readily controlled so that the connected cycle is truly automatic in its balance between heat required for feed heating purposes and heat available from exhaust and extracted steam. The same

cannot always be said for those cycles which employ steam-driven auxiliaries unless great care is used in the design and maintenance of the system.

Some designers believe that it is possible to improve on the cycle of Fig. 2 by omitting the supplementary turbine and substituting steam-driven auxiliaries. A typical cycle is illustrated in Fig. 3 which is a generic representation of the arrangement proposed for the over pressures. This permits the installation of a larger topping turbine than would be the case if the pumps were electrically driven. These boiler feed pump turbines will necessarily have a high water rate because of the small available pressure drop and care is necessary to insure adequate heat absorbing capacity in the water fed to the intermediate heater.

Another procedure which is followed at the West

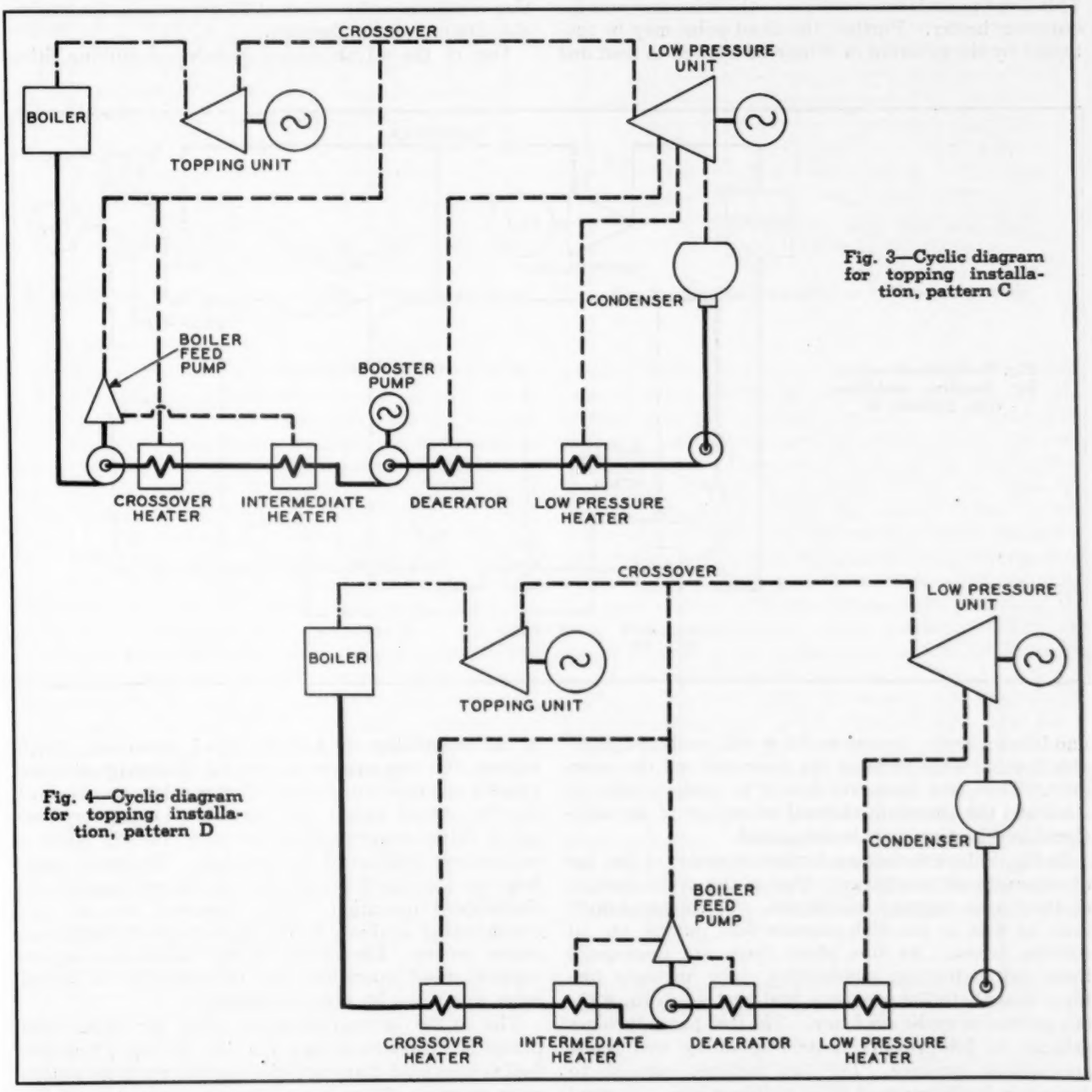

Fig. 3—Cyclic diagram for topping installation, pattern C

Fig. 4—Cyclic diagram for topping installation, pattern D

Springdale extension. The existing low-pressure end of the feed system is unaltered and the new high-pressure feed pumps have been superposed directly on the present plant. This procedure has much to recommend it in that there is the minimum disturbance to existing equipment. The high-pressure feed pumps are driven by steam turbines from the crossover and exhaust into a heater intermediate between atmospheric and the cross-

End and Rivesville Stations is diagrammatically shown in Fig. 4. Here the high-pressure boiler feed pumps are also driven by steam from the crossover. They exhaust to a heater, often conveniently the deaerator, which functions in the region of 215 to 250 F. The feed pumps take their water supply directly at this temperature and when compared to the plan of Fig. 3 have the advantage of handling colder water (250 F instead of

400 F). This saves on overall pump power and avoids the booster pump used in Fig. 3. The head required in Fig. 4, however, is the aggregate of that of the booster and feed pumps of Fig. 3, so that the horsepower of the former feed pump is greater than that of the latter feed pump alone. Consequently, a larger capacity driving unit is needed on this cycle and if the heat-absorbing capacity of the system is to be adapted to the feed turbine exhaust a bleed point is added to the feed pump turbine. This bleed point is intermediate to the deaerator and the crossover heater. Further, the bleed point may be controlled by the governor or it may be allowed to float and find its own level. In any event, it will result in a desirable heating stage between the deaerator and the crossover. Thus, the crossover heater is made smaller in itself and the attendant thermal advantage of the additional feed-heating stage is also gained.

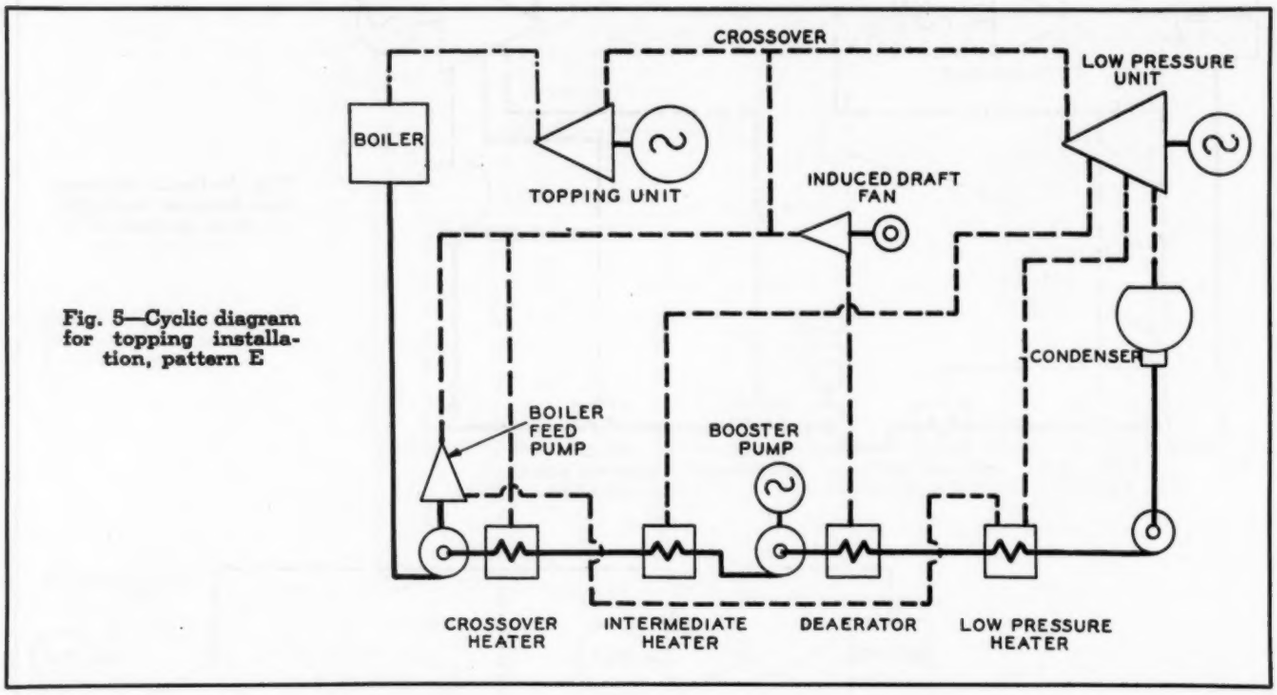

Fig. 5—Cyclic diagram for topping installation, pattern E

In Fig. 5 there is shown a further extension of the use of steam-driven auxiliaries. This is the cyclic pattern of the Logan topping installation. The induced-draft fans, as well as the high-pressure feed pumps, are all turbine driven. At this plant there are inadequate main unit extraction possibilities. The auxiliary turbines contribute the necessary feed heating steam without sacrifice of cyclic efficiency. The feed pump turbines exhaust to low-pressure heaters operating well below atmospheric pressure. The fan turbines exhaust to deaerators in the range of 20 to 40 lb per sq in. Control of heat absorbing capacity is effected by the supplemental use of main unit extraction which is automatic in its thermal flywheel action.

Figs. 1 to 5 are typical of some of the present topping practices and illustrate the extent to which steam and electric drives are being adapted to certain installations. The large size of these auxiliaries makes it possible to obtain turbines with an overall Rankine efficiency ratio sufficiently high as to be of interest from a thermal viewpoint. The horsepower requirements, together with the requisite running speeds, both contribute on this score. Fig. 6 shows typical Rankine efficiency ratio curves obtained on competitive boiler feed pump drives. These values which peak at 70 per cent overall, are entirely commensurate with the main unit efficiencies in the same operating pressure region particularly if generator, transformer, transmission and motor efficiencies are included. The state lines pass through the saturation condition and a value of 70 per cent is of considerable interest to the designer.

One of the advantageous features of turbine drive is the possiblility of variable-speed operation. With motors, this can only be accomplished directly at extra expense and then only a relatively few points are obtained on the control panel. For fans, dual-motor or two-speed drive, supplemented by vane control, offers a satisfactory solution of this problem. Hydraulic couplings are also much favored for the procurement of variable-speed operation. They, however, are not yet commercially available in the highest speed and largest power ranges. The turbine is well suited to complete variable-speed operation and consequently is looked upon with favor by many designers.

The utility of variable-speed drive for boiler feed pumps is illustrated in Figs. 7 to 10. In Fig. 7 is shown feed system head characteristics, against which the pumps must operate. They have been constructed from the basic superheater outlet pressure line A-A, which is to be maintained constant irrespective of load. To this has been added the allowances for pressure drop through the superheater, feed piping, feed heaters, check valves, feed regulators and the static elevational head to give the normal boiler feed pump discharge pressure B-B. This is the pressure when all equipment is operating as expected. For emergency conditions, however, there will be an extreme head requirement, C-C, needed to

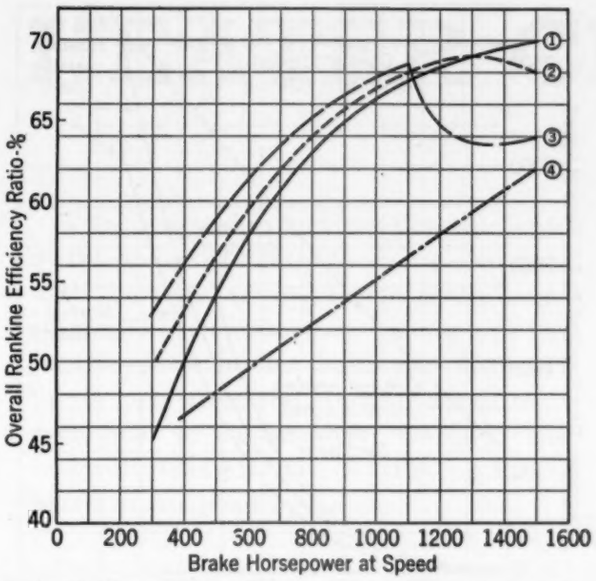

Fig. 6—Typical Rankine efficiencies for alternate auxiliary turbine designs

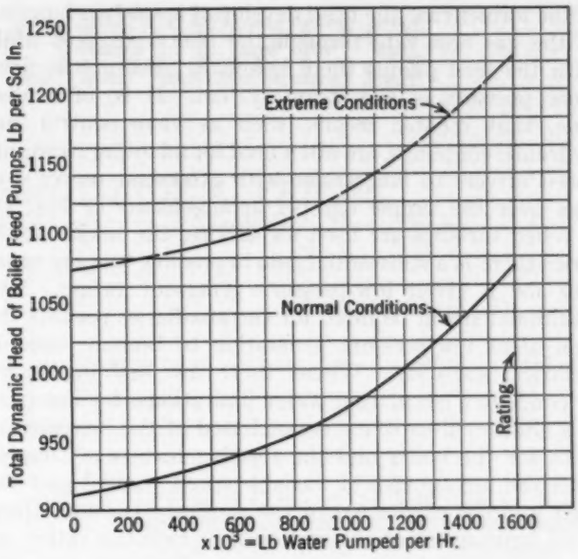

Fig. 8—Net head required on boiler-feed pumps

feed water to the boiler when the safety valves are blowing. The pumps normally will have to operate along the system characteristic B-B, but they will also have to meet the contingency of the extreme conditions represented by C-C. In the illustrative case shown there is a booster pump ahead of the main boiler feed pump which contributes the head D-D at the boiler feed pump suction. The net head, therefore, to be delivered by the main feed pumps is the difference between curves B-B and D-D, or C-C and D-D, for normal and extreme conditions, respectively. These differences are plotted in Fig. 8.

The characteristics of Fig. 9 are those obtained on the boiler feed pump to meet the service resistances constructed in Fig. 8 when the pumps are operated at constant speed. The excess head between the extreme and normal operating points at rating must be throttled out and thus represents a direct loss of energy. If variable-speed operation is resorted to the large power saving, reflected in the rule that power is proportional to the cube of the speed, can be largely realized. The entire gain cannot be effected because operation is not at the same efficiency point. However, a large gain can be drawn as is shown graphically in the curves of Fig. 10. Here the vertical distance between the constant-speed horsepower characteristic and the variable-speed characteristic shows a saving of as much as half the power required by the pumps. This gain by variable-speed operation is largely realized with the turbine drive because of the high Rankine efficiency ratios available. Even the higher Rankine ratios of the main unit cannot offset the gain from variable-speed operation.

Fig. 11 has been added to demonstrate the relative power requirements of the induced-draft fans for a typical large boiler. The upper curve gives the horsepower needed with constant speed operation and simple damper control. The lower curved lines reflect the wide change in power requirements with full variable-speed operation. These latter three curves pass through the origin and should be compared with the equivalent curves of Fig. 10 for the feed pumps. There is a larger possible gain with the fans than with the pumps because

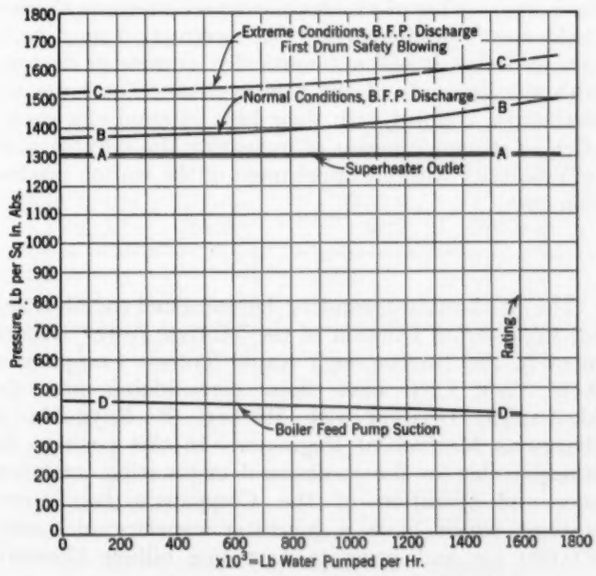

Fig. 7—Feed system head characteristics

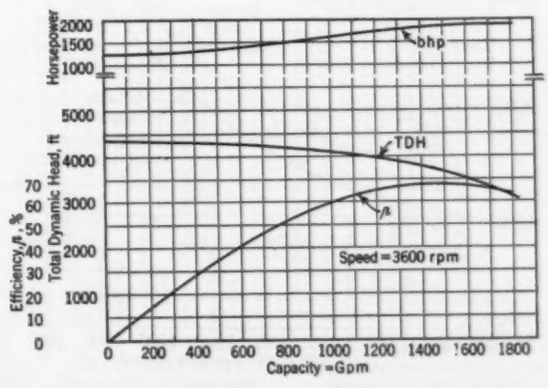

Fig. 9—Boiler-feed pump characteristics

in the former case the head developed is solely a function of the gas flow rate through the boiler passages while with the feed pumps there is always existent the fixed basic pressure of the steam system. It is, of course, true, that reliable means, such as vane control and hydraulic couplings, are often used for adapting constant-speed drivers to fan service with attendant power savings over the simple damper arrangements of Fig. 11.

When turbines are used for driving the major auxiliaries there is a substantial gain in possible topping capacity for a given low-pressure generator rating. The additional steam required for the auxiliaries permits the equivalent low-pressure generation to become saleable electrical capacity. Where there are building space or condenser circulating water limitations, for instance, this additional capacity is purchased at the incremental price for the boiler and the topping turbine. Coupled with the advantages of variable-speed control and the high internal efficiencies of the auxiliary turbines there is no impairment of the station heat rate but rather an improvement of this item. The additional capacity thus obtained is viewed by many designers as an economic factor of controlling importance.

The questions of maintenance and reliability of steam versus electric drives cannot be overlooked. With motors there is bound to be low maintenance, as long experience indicates. Reliability is, however, debatable on the old grounds of dependence on a source of electric supply or on a steam line. With topping plants, where crossover steam is used for drive purposes, this must include the adequacy of reducing valves and desuperheaters to meet the contingency of sudden outage of the high-pressure turbine. Nozzle and vane erosion on the auxiliary units might appear to be serious. A given depth of wear will be of far greater influence on the small passages of a 1000-hp turbine than on the larger passages of a 30,000-kw unit. Moisture is the prime cause of such wear. The quality of the steam, however, leaving the auxiliary turbine will be higher because of the higher back pressure. In the cases of the boiler feed pump turbines of Fig. 3, or the fan turbines of Fig. 5, the exhaust pressure is in the region of 50 to 100 lb per sq in. Here the expansion never crosses the saturation line so

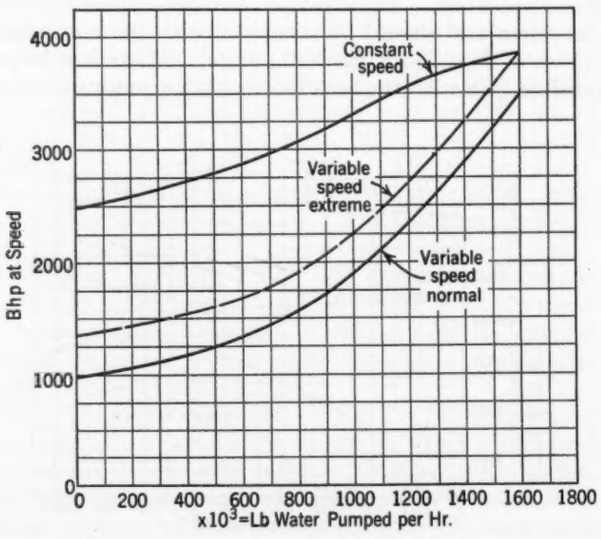

Fig. 10—Brake horsepower required on boiler-feed pumps

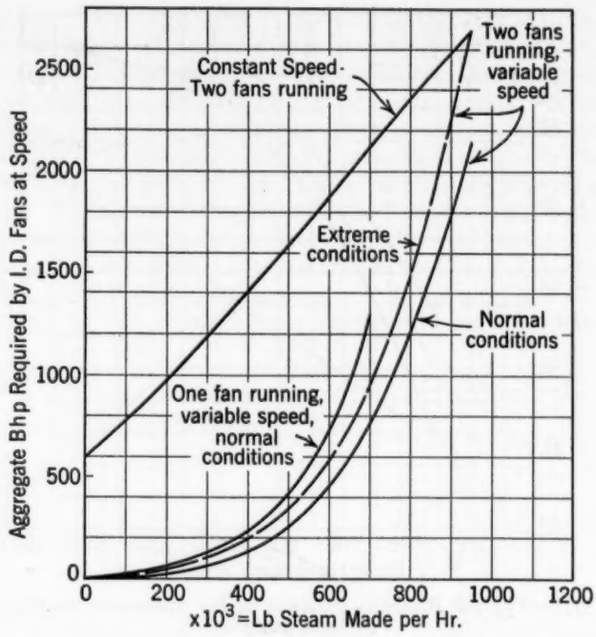

Fig. 11—Brake horsepower required on induced-draft fans

that the steam is continuously superheated on passage through the unit. Thus, the major cause of wear is eliminated, except for the brief starting-up periods, and good parts life is a reasonable expectancy.

The comments here given are an attempt to cite certain factors for and against the use of steam and electric auxiliary drives in the light of present practice on topping installations. The electric system has demonstrated its worth through past experience. Flexibility and reliability have been high and will continue to remain so. The cycle which operates exclusively on main unit extraction is completely automatic because the heat demands of the feed system are exactly matched at all times by the heat available from extracted steam. With steam auxiliaries the problem is more difficult in this respect, but it is far from incapable of solution. The cycle can be made automatic by the judicious use of such equipment in conjunction with main unit bleed or dual drive, either of which provides an adequate thermal cushion or flywheel effect. Such extraction must be so arranged that it will automatically increase or decrease with the demands of the feed system. The large size auxiliaries, coupled with their high internal efficiencies, offer an economic means of increasing the saleable main unit capacity without impairment of the station production cost.

The Mechanical Operating Division and the Mechanical Engineering Division of the Motive Power Department of the Interborough Rapid Transit Company of New York City have been consolidated into the Mechanical Division with **Herbert B. Reynolds** in charge as Mechanical Engineer. In this position he is responsible for the mechanical engineering, maintenance and operation of the Company's two power stations which have a generator capacity of nearly 400,000 kw and generate over one billion kilowatt-hours of electric power per year.

HEATER CLOSURES

By GEORGE A. FRICKE
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A discussion of progressive developments in the design of heater heads to keep pace with the trend toward higher steam pressures and the employment of larger units. In this development bolts and gasketed joints have given way, in the case of high pressures, to designs that transmit the hydraulic load on the cover directly to the barrel of the head by shear.

THE TREND to higher steam pressures and temperatures during the past decade has led to extensive research and development in both design and materials of construction, not only of boilers and prime movers but also of associated mechanical equipment such as feedwater heaters and various forms of heat exchangers. To withstand the higher pressures it was not always possible, nor desirable, to adapt low-pressure designs to the new conditions by merely making the structure heavier or by using a material having a higher allowable stress. A complete redesign of the structure to meet the new requirements was, in most cases, the only way to achieve an effective and acceptable design. There are here illustrated various stages in the development of closures for bleeder heaters and heat exchangers in keeping abreast with the high-pressure trend.[1]

In a broad sense it is possible to divide the pressure ranges into three groups when considering closure designs for such equipment. These may be taken as follows:

Low pressures, up to 300 lb per sq in.
Medium pressures, 300 to 900 lb per sq in.
High pressures, 900 to 2000 lb per sq in.

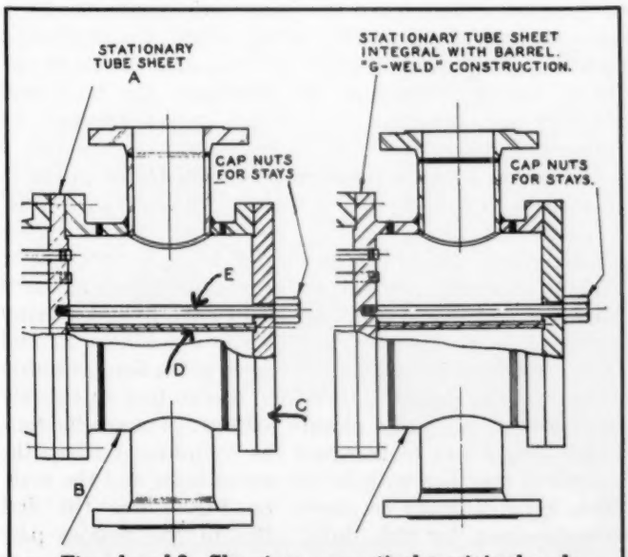
However, there are many exceptions since the real factor is the total load (hydraulic plus gasket bearing pressure) that must be taken care of in the bolting. It can be seen that a medium-pressure heater of large diameter would require a high-pressure type of closure since the total load on the cover would be as large as the load on the cover of a high-pressure heater of smaller diameter.

Fig. 1 illustrates one design for a low-pressure head. This head consisted of a tube sheet *A* into which the tubes are roller expanded, a water head *B*, a head cover *C*, a partition plate *D* and staybolts *E*. In making up this head it is necessary to maintain gasketed joints against the water pressure and employ staybolts to secure the gaskets under the partition plate. This design might be considered the basic head to which were applied the improvements in the development of heater closures.

With almost the first increase in sizes and working pressure the heater was redesigned as illustrated in Fig. 2. The head is essentially the same structure as that shown in Fig. 1 with the exception that it is made integral with the tube sheet, thereby eliminating one gasketed pressure joint between the tube sheet and water head. Incorporation of this feature reduced the number of gasketed joints that must be maintained in the head.

Maintaining the gasket under the partition plate was the next source of concern. The use of staybolts to accomplish this was not desirable because where the staybolts passed through the head cover cap-nuts were employed to avoid leakage of the water to the outside,

[1] Many of the described details are patented designs.



Figs. 1 and 2—Showing, respectively, original and redesigned low-pressure head

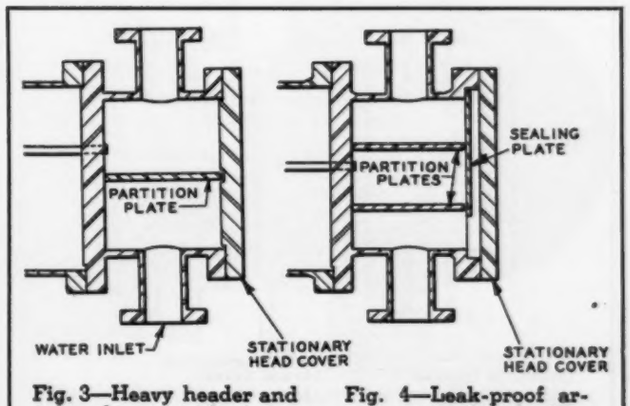


Fig. 3—Heavy header and tube sheet to eliminate staybolts

Fig. 4—Leak-proof arrangement by employing pass-seating plate

and they have often proved a nuisance. While it might not seem serious for a small quantity of water to leak past the partition plate, that is, bypass some of the tube surface in the unit, it nevertheless can at times greatly affect the performance of the unit as to the ultimate exit temperature of the water and the quantity of steam bled from the turbine.

The next development dealt with elimination of the staybolts by making the head cover and the tube sheet heavier so as to resist any deflections and thereby keep the gasket tight. This is illustrated in Fig. 3.

However, this scheme was not always positive in its action; hence, the pass-sealing plate was designed so as to make a positive leakproof arrangement. Fig. 4 illustrates this design. The partition plates are in no way attached to the cover but are welded to the tube sheet and the sealing plate is bolted to the barrel of the water head and to the partition plates. This structure can be of light construction because it is not subjected to the total hydraulic pressure—only the difference between the water pressure and the pressure drop caused by the water flowing through the tubes. The water enters the head and flows up in between the cover and the sealing plate, subjecting the cover to the full hydraulic load while the water flows through the tubes of the heater. The pressure drop of the water through the tubes assists the bolting in maintaining a tight gasket and thus prevents leakage of water from one pass to another.

The pressures which heaters were under up to this point in development were either below 450 lb per sq in. or, if higher, the units were relatively small in diameter and these designs were readily adaptable. With the advent of 600 lb heaters in large sizes it was found that

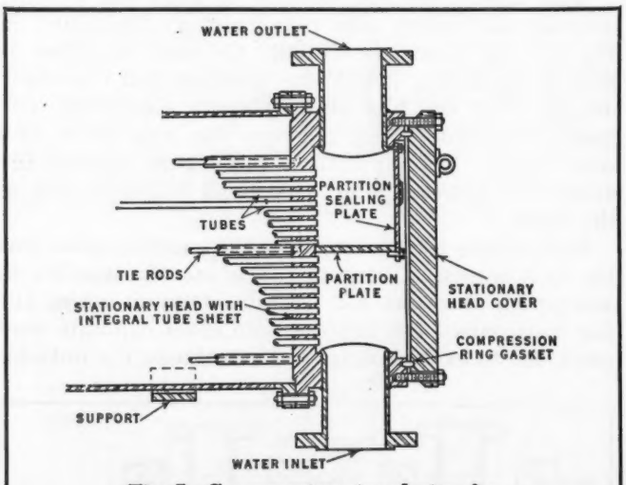


Fig. 5—Compression ring design for moderate pressures

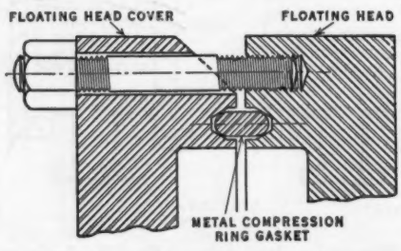


Fig. 6—Showing metal gasket used in compression ring design

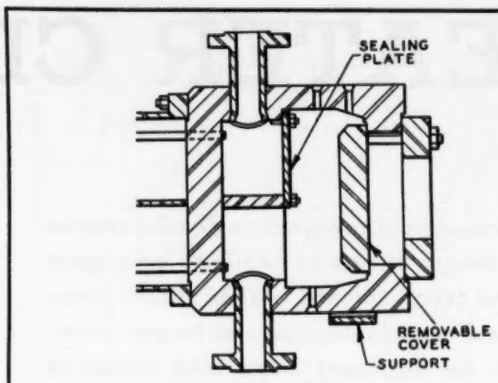


Fig. 7—Manhole type of cover

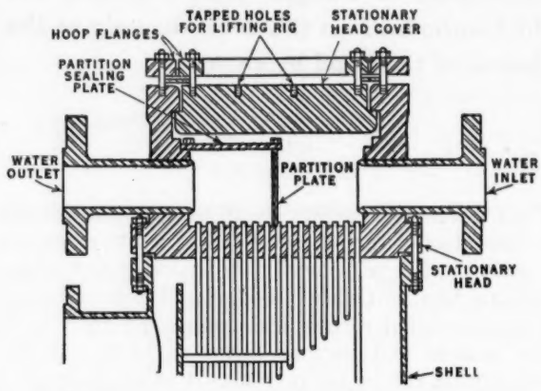


Fig. 8—Closure for vertical high-pressure heater employing breech-block principle

the size and number of bolts to accommodate the hydraulic load, plus the gasket bearing pressure, was becoming a problem. Therefore, the compression-ring, gasket-type of heater was next developed to handle this pressure, and also to avoid gasket trouble. Fig. 5 shows the general design of this heater and Fig. 6 a detail of the joint.

The gasket used for this joint is an accurately machined hoop of mild steel which is fitted into grooves machined into both the head flange and the head cover, thus providing a metal-to-metal bearing surface. The grooves are so machined that when the joint is tightened the hoop is put into compression; then, when the hydraulic working pressure is applied and the load taken in the bolts, thereby stretching the structure, the hoop will expand and maintain the proper gasket pressure to prevent leakage.

The next jump in pressure was from 600 to 1400 lb per sq in., together with a rather large increase in the diameter of the units. To meet these conditions an entirely new design had to be developed for the heads. While, as a rule, there are no objections to taking loads on bolts, the new conditions would have required many bolts of large diameter and it was a problem to find space for them and still keep the size of the flanges within reason. The problem, therefore, was to find an entirely new way of making a closure for high-pressure heaters. The answer was to transmit the hydraulic load to the barrel of the head without the use of bolts and the manhole type of cover as shown on Fig. 7 was the first development for this duty. Use of the sealing pass plate made possible the adoption of the manhole cover as a high-pressure heater closure. The hydraulic load was

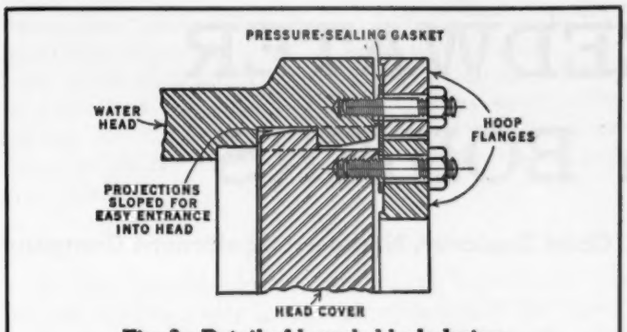


Fig. 9—Detail of breech-block design

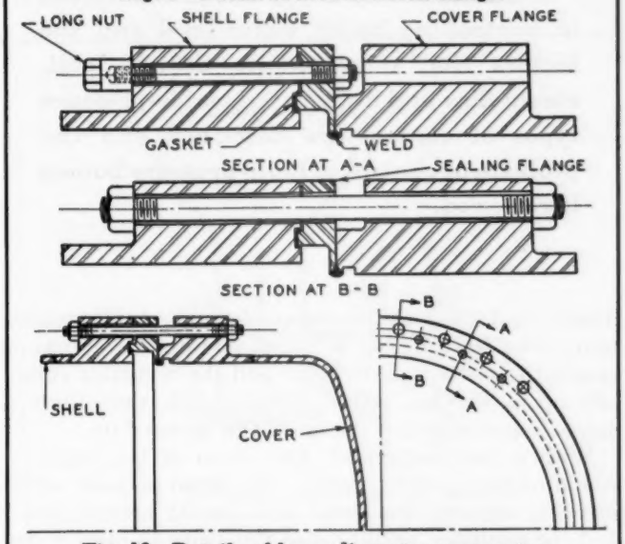


Fig. 10—Details of large diameter evaporator closure where load is taken on bolts

transmitted to the head by a shearing stress. However, this particular design was rather heavy and large for the diameter of the heater and a more practical way of accomplishing this was investigated.

One of the latest designs of high-pressure closure is the visible pressure seal which is shown in Fig. 8. In this the arrangement is somewhat similar to the breech-block design used extensively in modern ordnance and a further refinement of the manhole form of closure. A detail of the joint is shown in Fig. 9.

The head cover is a flat plate with machined lugs around the circumference which fit into place through notches machined into the flange of the heater head. The cover is then revolved about three inches, the width of a lug, thereby engaging the corresponding lugs on both the cover and the flange and the hydraulic load is transferred to the barrel of the head by shear. Sealing against leaks is accomplished on the outside by a visible pressure-sealing member requiring a relatively small number of bolts and which is easily accessible without removing the water-head cover. The sealing member consists of an inner and an outer hoop backing up a stainless steel ring which holds the metallic gasket in place. Since the bolts take care of only a small hydraulic load plus the gasket pressure they are under moderate stress and the joint can easily be kept leakless.

As an illustration of a high-pressure head where the hydraulic load is taken in bolts, Fig. 10 shows an evaporator door joint that has recently been developed. This head was designed for vessels having diameters up to 118 in. and for operation at medium pressures. However, the total load upon the door compares favorably with that of a smaller diameter high-pressure heater.

The feature of this construction is that the tightness of the gasketed joint is not dependent upon the tension in the large main cover bolts. A separate means, consisting of a sealing flange with small bolts and with an expansion member welded to the cover at the inner radius, provides the necessary pressure on the gasket to prevent leakage. This sealing flange carries practically no pressure load. The main bolts are proportioned to carry the pressure load.

In securing the cover to the shell, first the large main bolts are drawn up only "wrench tight" without excessive leverage, pull or sledging; then the small sealing flange bolts are drawn up "wrench tight" by means of the long nuts. Pressure in the shell will tend to elongate the large main bolts but the small bolts will hold the gasket tight.

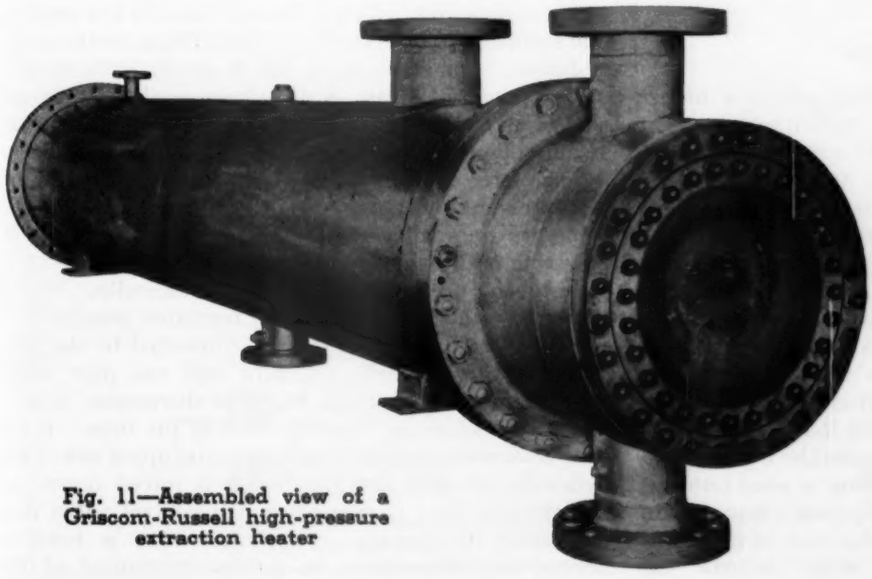


Fig. 11—Assembled view of a Griscom-Russell high-pressure extraction heater

Fig. 12—Head and cover for breech-block design showing projections for carrying the hydraulic load

Regulation of FEEDWATER LEVEL in BOILERS

By V. V. VEENSCHOTEN, Chief Engineer, Northern Equipment Company

A discussion of the principles involved in controlling boiler water level and the factors that influence it. Simple level, steam flow and differential water-pressure types of control are described and the problem of control of high-pressure boilers is reviewed.

THE TREND toward higher boiler ratings, higher furnace temperatures, smaller water storage spaces and more rapidly fluctuating loads calls for sensitive water level adjustment. Moreover, modern boilers have proportionately less steam liberating area with reference to the volume of water contained. This is particularly true at pressures between 400 and 700 lb. At higher pressures swelling or shrinking of the water level is less pronounced because the densities of the steam and the water more closely approach each other.

Among the factors that influence boiler water level are:

1. Speed of load changes, influenced by the ratio of heating surface to water volume
2. Volume of steam held in the water at any given instant, as influenced by the circulation
3. Ratio of steam liberating area to water volume
4. Boiler drum pressure
5. Feedwater temperature
6. Rate of steam output
7. Rate of feedwater input
8. Location of the water column
9. Concentration of the boiler water

With every feedwater regulator the rate of water input is responsive to certain influences. Where the response is only to changes in boiler water level, the regulator is of the "simple level control" type. This is widely used. However, where the primary response is to changes in the rate of steam flow a "steam flow" type of regulator is employed. Use of this type is increasing. Both types will be explained.

Simple Level Control

A "simple level control" regulator carries a higher water level on light loads than on higher ratings. On steady loads, the water level is held constant by a constant rate of feed proportioned to the load and with gradual load changes the rate of feed is gradually adjusted. However, on sudden increase of load the feed is temporarily reduced, whereas on sudden drops in load it is temporarily increased. The reason for this is as follows:

On a sudden load increase, there is at first a slight decrease in drum pressure and the steam bubbles in the boiler water expand. This causes the water level to rise and the regulator reacts by decreasing the valve opening. This may seem wrong but it is exactly what should happen. The water input should be decreased so that the heat energy in the boiler can be used immediately to change as much water as possible into steam. If the feed were increased, the sudden rush of comparatively cold water into the drum would decrease the number and size of the steam bubbles and retard evaporation. In a few minutes the increased rate of evaporation causes the water level to lower and the regulator gradually opens the feed valve. During this time there is ample opportunity for the fires to be speeded up.

With a decreasing load, the action of the regulator would be exactly the reverse. The drum pressure would increase slightly, the water level would become lower and the regulator would cause the valve to open wider. The increase in feed would absorb heat from the boiler and during the time the fire was retarded the flow of water would be gradually reduced. A higher water level is carried for each sub-normal load and at no load the level is at the highest permissible point and the valve closed.

Figs. 1, 2 and 3 illustrate units of the simple level control type.

Control for High-Pressure Boilers

Contrary to common belief, it is easier to stabilize boiler water level at pressures of 1000 lb and over than at lower pressures. This is because there is less swelling and shrinking to be taken into consideration, as the steam is denser. For instance, at 230 lb pressure the volume of one pound of steam is slightly more than 100 times that of an equal weight of water, whereas at 1400 lb the steam volume is only 12.7 times that of the water.

Therefore, the simple level control is adequate for high-pressure boiler feeding, although where high water velocities are involved the equipment must be still more powerful and rugged. One way to secure this is to operate the feed-control valve hydraulically. Fig. 4 shows how this is done with the regulator illustrated in Fig. 3. The thermostat lever is connected to the pilot valve of the hydraulic operator and the pilot valve remains in neutral except when the thermostat lever is moved by expansion or contraction of the tube. If the tube contracts, liquid is admitted to the upper end of the hydraulic cylinder and the piston is forced down. A straight-line force is exerted on the control valve thus decreasing its opening. The pilot valve is reset in neutral and there must be further movement of the

thermostat before the valve can move. If the thermostat tube expands, liquid is admitted into the lower end of the hydraulic cylinder. The piston is forced up and exerts a straight-line force increasing the opening of the valve. Thus there is ample power to overcome friction and the unbalanced forces created by the high velocities.

Steam Flow Type of Control

Boilers handling rapid, wide load fluctuations are commonly fitted with feedwater regulators of the "steam flow" type. These usually carry a higher water level on heavy loads than on light loads, thus permitting closer stabilization of the water level where swelling and shrinking on sudden load changes are pronounced.

Steam-flow type regulators are necessarily multiple-element controls. In addition to the control element corresponding to the changes in rate of steam flow, there must be one or more other elements influencing the opening of the feed control valve. Some such regulators are influenced by three elements, one of which responds to steam flow, one to water flow and one to boiler water level. The steam flow element is subjected to the pressure drop between the boiler drum and the superheater outlet; the water flow element by the pressure drop between the economizer inlet and the boiler drum; and the boiler water level is held within the desired limits by the thermostat which functions much as a simple level control. This type is known as a three-element control.

A simpler and less expensive type of steam-flow unit is the two-element or "double-control" regulator which is responsive to the steam flow and to boiler water level

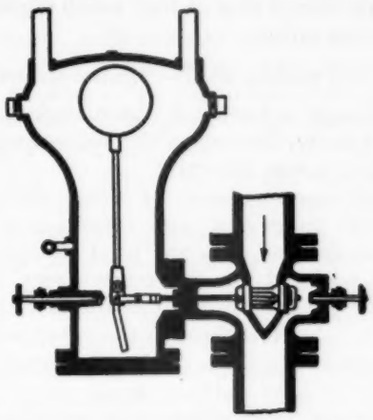


Fig. 1—Float type feedwater regulator

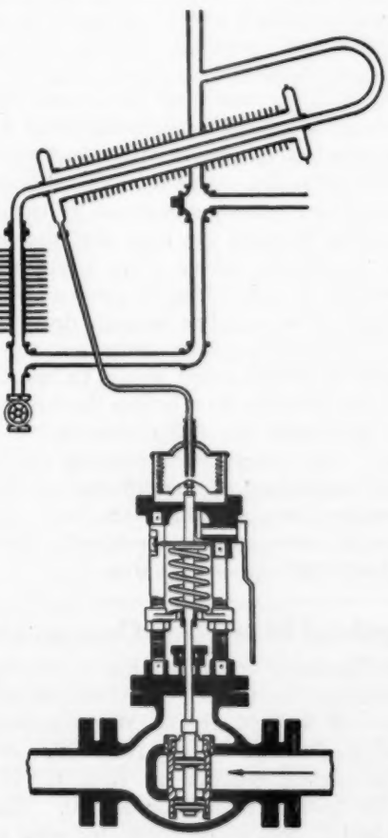


Fig. 2—Thermo-hydraulic diaphragm type of feedwater regulator

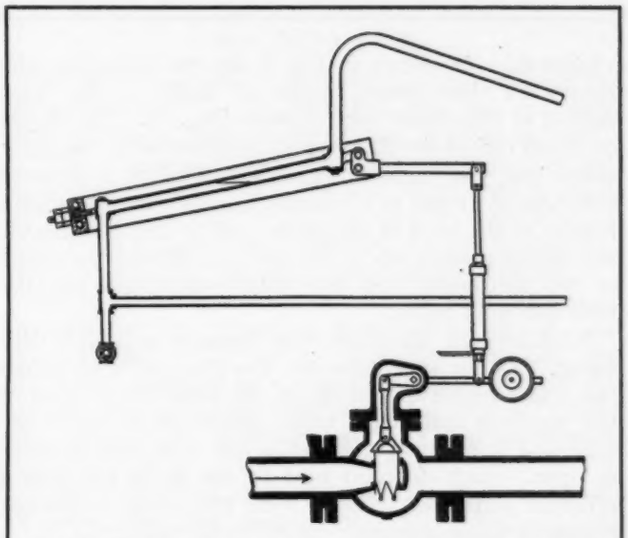


Fig. 3—Thermostatic expansion-tube type of regulator

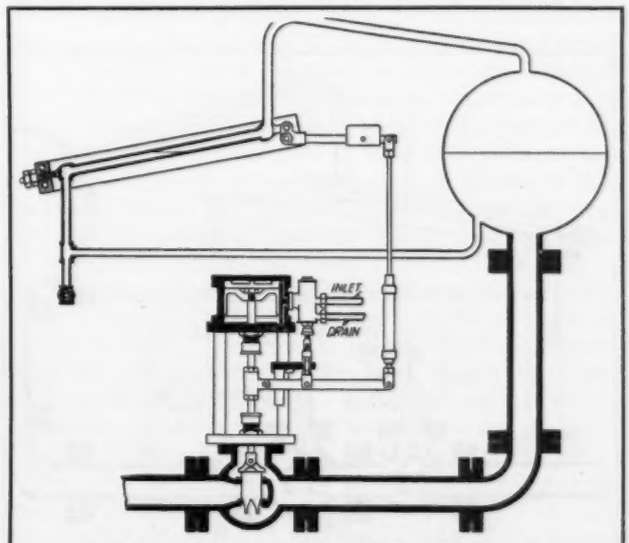


Fig. 4—Hydraulically-operated regulator for high-pressure boilers

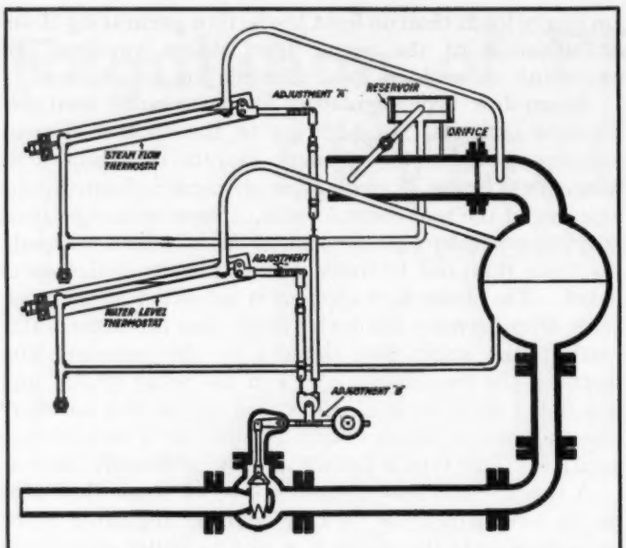


Fig. 5—Two-element, double-control regulator

variations. As shown in Fig. 5 the two actuating elements are thermostats similar in design to the thermostat of the simple level control (Fig. 3). The upper or steam end of the thermostat is connected to the main steam line, just ahead of an orifice. The lower or water connection is made to a reservoir connected to the steam header at the level of the upper end of the thermostat and on the trailing side of the orifice. Steam condensing in the thermostat fills the water connection and the reservoir with water.

With no load the steam flow thermostat is filled with water. As the load increases, the pressure drop across the orifice increases and forces the water level down in the reservoir and in the tube. More steam enters the tube and it expands, permitting the feed control valve to open. Each different load on the boiler provides a different water level in the tube and thus a different degree of valve opening.

If there is no change in the boiler water level, the valve is influenced by the steam flow thermostat which increases the feed as the steam flow increases. On the other hand, if the water level changes, the valve is also influenced by the water level thermostat which decreases the feed as the level rises, and increases the feed as the level lowers. The result is that the influence of each control element tends to balance the other on sudden changes in load.

With a sudden increase in load the steam-flow thermostat instantly acts to open the valve. Concurrently, the water level thermostat, reacting to the swell, acts to close the valve. The control valve does not move and the rate of feed remains the same until the more rapid rate of evaporation has reduced the volume of the boiler water enough to offset the increase in volume caused by the swell. The feed value then opens sufficiently to produce a rate of feed which equals the steam output at that rating.

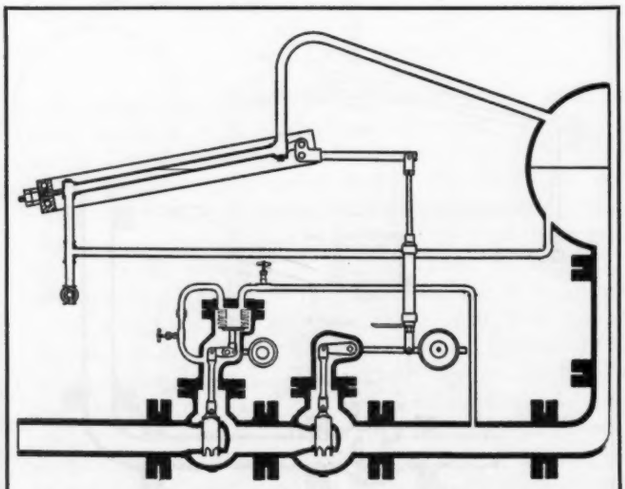


Fig. 6—Differential valve in series with feedwater regulator valve

Differential Water-Pressure Control

Flow through a regulator valve depends upon two variables, namely, the area of the valve opening and the pressure drop across the valve.

Any feedwater regulator, of either the simple level control or the steam flow type, is influenced by changes in boiler water level. As the level changes, the valve opening is adjusted for the correct rate of feed. If, however, the pressure drop across the valve is allowed to vary appreciably, flow through the valve will vary even with a constant opening. This will eventually change the boiler-water level and thus change the valve opening, but the flow variations will have an undesirable effect on the boiler load. Flow should be varied only as needed for load changes. Even small changes in feed-line pressure may double or triple the pressure drop across the feedwater regulator valve. Hence it is advisable to employ differential pressure control where the excess water pressure, in normal operation, will vary more than plus or minus 25 per cent from the desired value.

Fig. 6 shows a schematic installation of a differential pressure control valve in series with a feed water-regulator valve. The actuating element of this differential valve is a brass sylphon bellows influenced by the difference in water pressure between the inlet and the outlet of the feedwater regulator valve. The counterweight (the valve may also be spring-loaded) gives a positive loading for the desired differential or pressure drop.

If the differential pressure decreases, the differential valve opens to admit more water to the feed control valve. If the pressure drop across the feedwater regulator valve increases, the differential valve admits less water to it. Any excess water pressure over the desired differential is absorbed by the differential valve.

To conserve space and make the unit more compact the differential valve may be combined in the same body with the feedwater-regulator valve.

Strength of Materials Course at M.I.T.

A special summer school course on Strength of Materials, concluded by two all-day Conferences on Fatigue and Creep, will be held at the Massachusetts Institute of Technology June 21 to July 16, 1937. All intending to take part should register by June 1. For complete program the tuition fee will be $80.00. The Graduate School of M.I.T. will allow credit for nine units of advanced study. For further particulars address Prof. John M. Lessells, M.I.T., Cambridge, Mass.

How Engineers May Profit From Patented Inventions

By LEO T. PARKER, Attorney at Law, Cincinnati, Ohio

MANY later decisions of the higher courts vary or contradict old patent law, and engineers must necessarily be informed of the modern interpretations of the law in order to be able to enter successfully the patent field.

For illustration, until recently the old law was established that an engineer was, under all conditions, the sole owner of all patents of his inventions although he may have perfected the inventions by experiments made on the employer's time, and with the latter's tools and equipment (*229 F. 150*). In an old decision of the U. S. Supreme Court (*149 U. S. 315*), it was held that when an employer hires a skilled workman to work in or take charge of his plant and the employee devotes his time to making improvements, he and *not* the employer is entitled to ownership of the patents obtained for the inventions.

The legal effect of this old law was that although an engineer was paid to improve his employer's business and perfect improvements the engineer owned all patents obtained for inventions that he perfected while in the employer's hire. In other words, the employer merely held a "shopright," which gave him the right to make, sell and use the inventions, but the engineer was privileged to make, sell and use the invention and license it to competitors, because he actually owned the patents.

Interpretation of Old Law

It is interesting to know that in the recent case of Standard Parts v. Peck (*264 U. S. 53*) the U. S. Supreme Court rendered a decision in which interpretation of this previously established law was changed so that now an engineer who perfects an invention in the employer's plant, and while being paid by his employer, has no legal interest in the patent. Therefore, at present an engineer who is hired to improve or perfect his employer's equipment and obtains a patent on an invention can gain nothing by contending that the employer's only interest in the patent consists in a "shopright" to make, sell and use the invention. In fact, the employee has absolutely no interest in the invention unless he proves that he perfected the invention while off duty or while he was not being paid by his employer. Obviously, if the employee perfects the invention at his home, the employer has no interest in the invention.

Therefore, it is quite apparent that engineers who are not informed regarding the present law may labor at considerable expense, time and effort to perfect an invention, but if he experiments in his employer's plant, or wholly or partly perfects the invention on his employer's time, the employer owns the invention and patent.

Agreement to Sign

In order to secure exclusive ownership of patents obtained by engineers, the employer must include in the contract of employment a clause by which the employee agrees to assign all patents obtained while in the employment. Under a contract of this nature the employer may claim ownership to all inventions of the employee, although the latter perfected them at his home or while off duty. However, an executive or engineer who has not agreed to assign patents to his employer may claim ownership of patents he invents while at home or off duty, but he cannot use his position to obtain valuable ideas from employees and then file applications for patents in his own name. (*See 187 Fed. 840.*)

Rights of User Varied

During the past few years the higher courts have established a ruling to the effect that any firm or person who purchases a patented device may use it *as he sees fit*. In other words, the seller of a patented device cannot under the new and modern law *restrict* the uses to which the purchaser of the article may put it. Therefore, a notification attached to or imprinted upon a patented device intended to limit or restrict its uses is illegal.

Many sellers of patented equipment have attempted to avoid this law by selling patented equipment conditionally, and under special contracts. However, the courts look through any obscurity intended to avoid the established law that the purchaser of a patented article may use it as he desires and without any interference from the seller.

The higher courts have held that a contract is an *outright sale* if the seller attempts to retain control or future uses of the patented device.

However, if an engineer desires he may *refuse* to sell and permit others to use his invention. In one case (*47 U. S. 146*) the Supreme Court of the United States held that if a patentee sees fit, he may reserve to himself the exclusive use of his invention or discovery, although the grant of a patent is made upon the expectation that he will either put his invention to practical use or permit others to avail themselves of it upon reasonable terms. However, his right is exclusive, and so clearly within the constitutional provisions in respect of private property that he is neither bound to use his discovery himself nor permit nor license others to use it.

Of course, any plant owner who uses patented equipment or apparatus under a license is bound to abide by the exact terms of the license contract. In a license contract the patentee usually agrees to permit the other party to use the invention for a specified period and for stipulated purposes. The patentee receives payment from the licensee in consideration of the rights granted.

Yet a license contract may be invalid and illegal for many reasons, and in one recent case (*226 U. S. 20*) the Supreme Court of the United States held that a license agreement is void by the terms of which a patentee endeavors to eliminate competition.

In another recent case (*258 U. S. 451*), the court held a license contract void because it contained a stipulation by which the user of patented equipment agreed not to use devices made and sold by competitive manufacturers. Moreover, lease contracts are void which merely are disguised contracts of sale. For illustration, one court held that a contract is void by which a purchaser agreed to pay the seller certain amounts in installments with the privilege of purchasing the device for an insignificant amount after the installments were fully paid. A contract of this nature is held by modern courts to be legally a sale although its terms are intended to change the legal status of the parties.

When Infringement Exists

Of course, no one who uses patented equipment without authority of the patentee can avoid liability in damages if he refuses either to purchase patented equipment outright, or license it under terms designated by the patentee. In other words, if the prospective user refuses to contract according to the terms of the patentee no matter how unreasonable his terms appear to be, the prospective user cannot in any manner gain an opportunity to use the patented equipment until the seventeen-year term of the patent expires.

A very common source of legal controversy arises where two or more persons, by separate acts, attempt to obtain the benefits from a patented article without authority of the inventor. When the acts of two or more persons finally effect infringement *all* of those involved are guilty of *contributory infringement* and are liable for the total profits earned, plus the amount of damages incurred by the patentee. Furthermore, where the infringement is willful, the court may render a verdict in favor of the patentee for *three* times the profits earned by infringers, plus *three* times the damages incurred by the patentee.

Repairs Permissible

Another common source of litigation arises when a purchaser desires to keep indefinitely in operation and service a patented machine by supplying new parts as the old ones wear out. Under all circumstances the purchaser of patented equipment must not, unless he invites a suit for infringement make such repairs that are equivalent to rebuilding a patented device.

The higher courts hold consistently that when a patented machine wears out, the user must discard it. He cannot make it over by supplying new parts and repairs. Of course, this does not mean that a user cannot supply a broken part, but this law is applicable only to worn out parts.

Kinds of License Agreements

Any license agreement is valid, providing it does not relate to an illegal transaction which, of course, is rare with respect to production and distribution of power, steam, refrigeration, compressed air and use of boilers, turbines and motors.

A license agreement is valid which, for instance, limits the use to which the user may put the patented device or equipment. Or, a license agreement is valid by the terms of which a patentee grants privileges for a limited period.

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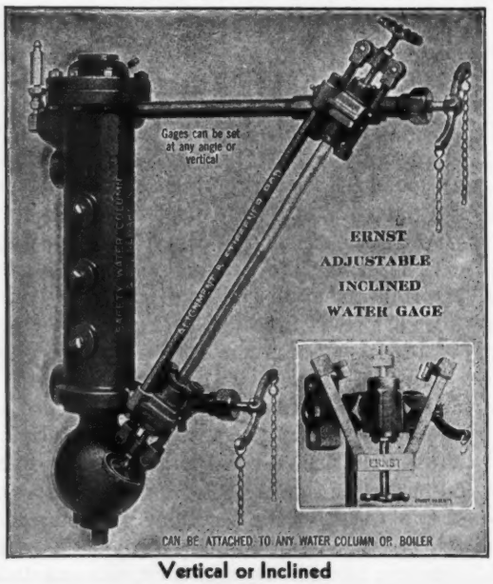
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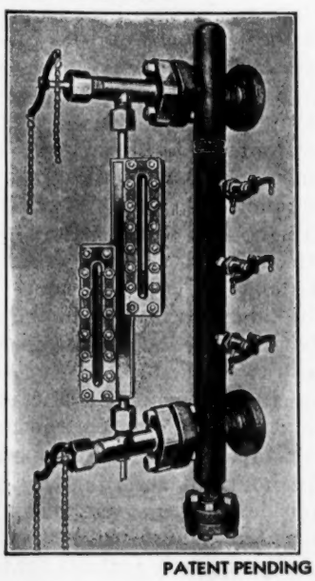
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As reported in the foreign technical press

Filming Combustion of Pulverized Coal

Examination of the combustion of pulverized coal by motion photomicrography, as a means of studying the performance of different types of burners, is discussed by A. G. Arend in the January issue of *The Fuel Economist* (London).

Based on experience in taking motion photomicrographs of rapidly moving protozoa, a micro camera with ocular attachment was devised for the filming of ignited coal particles in their different stages. Only a small amount of magnification is necessary and the movement of the particles in bulk is followed by the eye through a "cooler" and by means of a "beam splitter" whereby part of the light comes through the eye-piece for visual inspection while the film is being taken. This avoids taking a speculative exposure. The light passes through a filter before reaching either the eye or the film.

Selection of the proper filter has been found most important to the success of reproduction, as is also the exposure ratio. If the filters used cause a reproduction showing too much contrast, a loss of detail may result and the variations in the size of coal particles will not be clearly revealed. Dark green glass has been found much better than blue glass, which is not satisfactory in taking films. The speed of the film is varied to suit the rate of combustion and the speed of the burning particles.

Freedom from vibration is also essential and the apparatus is suspended from a specially constructed stage to avoid the effects of vibration from the burner.

Heating with Superheated Water

An interesting system of central heating is employed for a group of new Cantonal Administration buildings at Zurich, Switzerland. This, as described in the last quarterly of *Sulzer Technical Review*, employs "superheated water" heated in a vertical spirally-welded tank, or accumulator, by the condensation of steam at 140 lb. In winter this steam is extracted from a steam turbine and in summer it is produced by an electric boiler.

The steam at 140 lb is introduced into the upper part of the accumulator. Here it is condensed by water introduced into the steam space and flowing over baffle plates. This water is taken from the lower part of the vessel by a circulating pump. By this means the water in the upper part of the accumulator is gradually brought up to around 340 F at which temperature it is pumped into the heating mains. The return lines enter the lower part of the accumulator. When the quantity of heat introduced is greater than that withdrawn the level between the hot and the colder water is lowered and the accumulator is being charged. Inasmuch as the supply of extracted steam varies with the load on the turbine, the accumulator is able to supply a steady heating demand. Automatic regulation is employed for the steam flowing to the accumulator.

Tube Changes through Bending

In order to check the design of superheater coils for a high-pressure boiler a number of tests were carried out to establish the effect of different bending radii, the results being discussed by M. Ulrich and K. Wellinger in the December issue of *Mitteilungen*.

The tubes were of chrome-molybdenum steel, 38-mm outside diameter, and had a wall thickness of 5 mm.

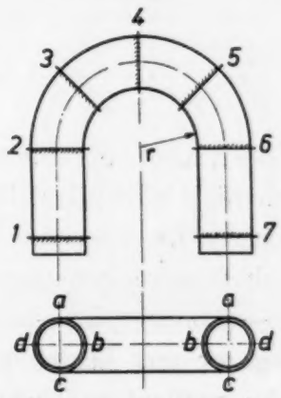

Fig. 1—Indicating locations at which measurements were taken

The material had a tensile strength of 65,000 to 78,000 lb per sq in. and an elongation of 20 per cent.

Cold bending was carried out with radii of 0.5, 1.5, 2.1, 2.5 and 2.9 tube diameters and hot bending with 0.5 tube diameter. Measurements were made of deviations of outside diameter from the original tube diameter, the state of unroundness, changes in wall thickness and the shape and change in free cross-section.

Fig. 1 shows schematically a bend of radius r, the measurements being taken at the sections 1 to 7. The abscissae for the curves in Fig. 2 represent these points of measurement and the ordinates the deviations in millimeters of the outside diameter normal to the plane of bending. The upper set of curves refers to the cold bending tests at five different radii and the lower set represents both hot and cold bending at a radius equal to one-half the diameter. The dotted lines represent deviations from the original diameter in a plane parallel to that of the bending.

For a tube with bending radius up to three times the nominal diameter the unroundness was found to be 4 to 5 per cent. According to the rules of the German Asso-

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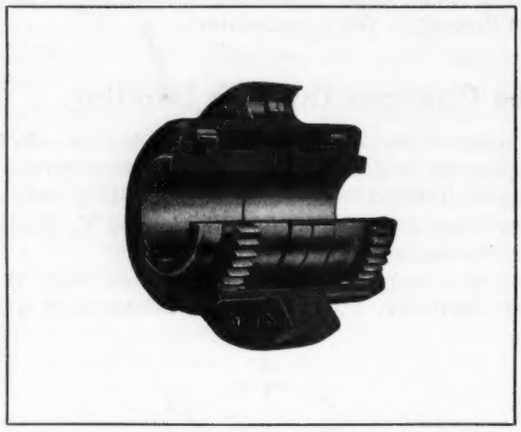

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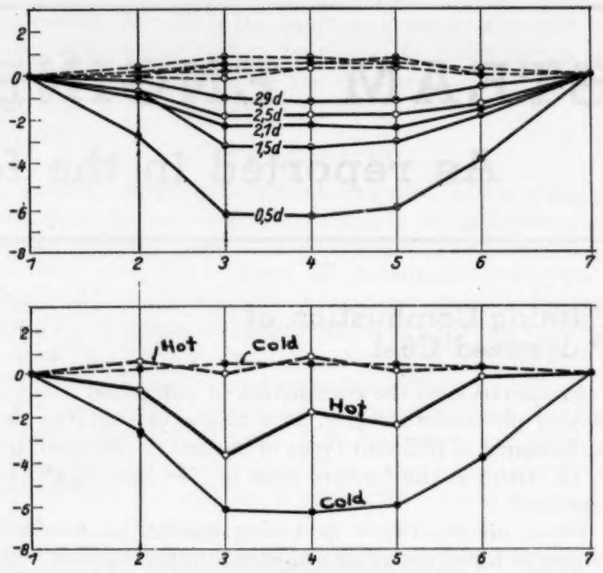

Fig. 2—Upper set of curves shows deviations in outside diameter for different bending radii with cold bending. Lower set of curves compares deviations for cold and hot bending with radius of one-half the tube diameter

ciation of Large Boiler Owners, for high-pressure boilers the unroundness shall not exceed 12 per cent and this limit was found to be exceeded only with a tube bent cold to a radius of one-half the diameter.

Change in Wall Thickness

Fig. 3 represents a section through a tube bent cold to a radius of one-half the diameter.

Referring to Fig. 1, it appeared that at points *a* and *c* the wall thickness was not materially changed, the greatest changes having taken place, as would be expected, at the inside and outside of the bend, or at points *b* and *d*. For a bending radius of one-half diameter the thickening of the metal on the inside of the bend is the same for both hot and cold bending, but for hot bending the decrease in thickness is more apparent on the outside. The greatest decrease in wall thickness takes place, not in the middle of the bend, but at both sides of the bend.

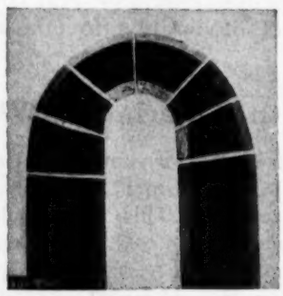

Fig. 3—Section through tube bent cold to a radius of one-half diameter

As to deformations of cross-section it was found that all were somewhat flattened at the outside of the bend, and the decrease in free area of the section grew rather steadily up to 21 per cent with a decrease in bending radius.

"Top" for British Industrial Plant

A "topping" installation has recently been applied to a central industrial power plant in England, as described in the January issue of *The Steam Engineer*. This plant furnishes electricity, water and steam for process and heating to the Slough Trading Estates, Ltd.—a community of various manufacturing industries that has been in existence for the past eighteen years.

The old boiler plant supplied steam at 200 lb and 600 F to five condensing turbine-generators, aggregating 13,500 kw capacity, and live steam at this pressure for heating and process. The present extension consists of two modern stoker-fired boilers, equipped with economizers and air heaters, and supplying steam at 600 lb, 800 F to a 2000-kw back-pressure turbine which exhausts to the 200-lb steam mains. At present the quantity of this exhaust steam is capable of supplying only 3600 kw of low-pressure capacity but with the installation of additional high-pressure boilers to replace two pressure boilers all the power and steam demands will be met from this source and the efficiency of the plant will be further improved.

Superheat Control

Control of superheat at the new Fulham Power Station in England is effected by means of desuperheaters. Two of these are installed on each boiler between the primary and secondary superheaters. After passing through the desuperheater the steam is raised by the secondary superheater to 850 F, thermostatic control being employed to maintain the final temperature within close limits. The desuperheaters are 30 in. diameter with forged-steel bodies and carbon-molybdenum cast-steel headers for 12-in. steam branches.—*World Power* (London)

New Turbines for Japan

Japan, which has already developed its water power to a considerable extent, is now engaged in adding to the capacity of its steam plants. As described in the January 1937 issue of *The Far Eastern Review*, a 50,000-kw steam turbine-generator has recently been installed at the Tsurumi Station of the Tokyo Dento K.K. This is a two-cylinder, 1800-rpm machine having a 3000-kw direct-connected house turbine. A 35,000-kw unit has been added to the Nagoya Station of the Toho Electric Power Company which supplies the industrial districts of Kansai and Kyushu, and the Taiwan Electric Power Company has ordered a 38,000-kw unit. This last unit will run in parallel with a hydroelectric system which, when there is ample water available, supplies the whole load demand. At such times the generator of the steam unit will operate as a synchronous condenser for power factor correction and for such purpose can be uncoupled from the turbine.

All three units are of German manufacture, being supplied by AEG.

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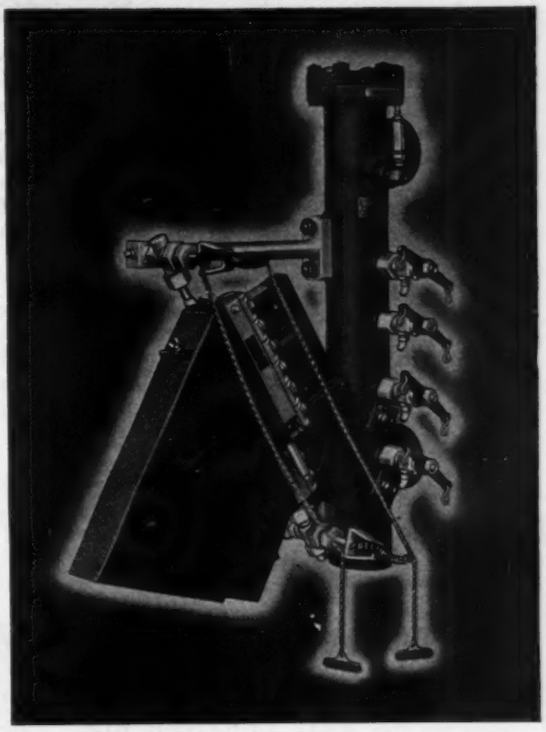

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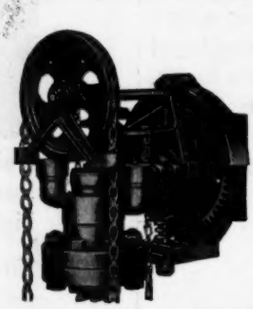

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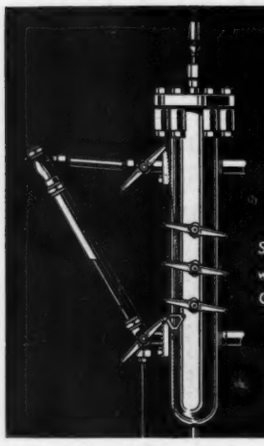

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